COMPUTER-AIDED INTERACTIVE, GRAPHICAL DESIGN OF MULTI-SPEED MACHINE TOOL GEARBOX

by

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by

ASHOK GUPTA

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DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY KANPUR

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CERTIFICATE

Certified that the work contained in this thesis, entitled 'COMPUTER-AIDED INTERACTIVE, GRAPHICAL DESIGN OF MULTI-SPEED MACHINE TOOL GEARBOX' has been carried out by Ashok Gupta under my supervision and that the same has not been submitted elsewhere for a degree.

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NOMENCLATURE

a_{ij}	=	Center distance between j^{th} gear pair of the i^{th} stage
a_c	=	Corrected center distance
a_m	=	Average center distance
b	=	Face width in mm.
B	=	Spindle speed range
C_i	=	Number of discontinuities on the i^{th} shaft
d	=	Pitch circle diameter
d_K	=	Tip circle diameter
d_g	=	Base circle diameter
d_f	=	Root circle diameter
d_b	=	Rolling circle diameter
d_r	=	Roller diameter
GR	=	Gear ratio
M_i	=	Over roller reading
m_{ij}	=	Normal module of the j^{th} gear pair of the i^{th} stage
N_s	=	Number of groups or stages
N_h	=	Total number of shafts
NS_i	=	Total number of speeds on the i^{th} shaft
NT_i	=	Number of transmission ratios in the i^{th} stage

NP_i = Number of pitch line velocities in the i^{th} stage

p_i = Number of transmissions from i^{th} stage

P = Horse power of the motor in KW.

R_i = Speed step ratio of i^{th} shaft

SP_m = Speed of input motor

S_i = Magnitude of discontinuity on the i^{th} shaft

SL_i = Lowest speed on the i^{th} shaft

SP_{ij} = j^{th} ideal speed of the i^{th} shaft

SP_{mij} = j^{th} modified speed of the i^{th} shaft

U_1 = Largest transmission ratio permissible

U_{ij} = j^{th} transmission ratio in the i^{th} stage

U_{mij} = j^{th} modified transmission ratio in the i^{th} stage

V_{ij} = j^{th} pitch line velocity within the i^{th} stage in m/sec.

x_i = Group or speed characteristic of the i^{th} shaft

X_1 = Profile shift factor for the driver gear

X_2 = Profile shift factor for the driven gear

y_i = Order of discontinuity on the i^{th} shaft

Z = Number of spindle speeds

ZMIN = Minimum number of teeth permissible

Z_{1ij} = Number of teeth on the j^{th} driver gear of the i^{th} stage

Z_{2ij} = Number of teeth on the j^{th} driven gear of i^{th} stage

Z_{v1} = Equivalent number of teeth on driver gear

Z_{v2}	=	Equivalent number of teeth on driven gear
Z1,Z2	=	Temporary number of teeth on driver and driven gears
α_{on}	=	Pressure angle on pitch circle (Normal section)
α_{o}	=	Pressure angle on pitch circle (Transverse section)
α_{b}	=	Pressure angle on rolling circle (Transverse section)
β_{ij}	=	Helix angle on pitch circle of the j^{th} gear pair of the i^{th} stage
β_{g}	=	Helix angle on base circle
$\emptyset$	=	Spindle speed step ratio
η	=	Transmission efficiency

ABSTRACT

In the present work, a methodology of interactive, graphical design of multi-speed machine tool gearbox has been developed. Based on the proposed methodology a computer program for interactive design has also been developed. The overall design process covers the design of the kinematic structure of the gearbox, the design of the speeds of the shafts as well as the number of teeth on various gears, face width calculation and generation of the inspection data. It is possible in the present approach to take into account several alternative kinematic structures of crossed, open or mixed type. It is also possible to take into account discontinuities in the spectrums of speeds on intermediate shafts and also the introduction of passive shafts. The proposed approach has been illustrated with the help of three case studies and the results obtained have been discussed.

CHAPTER-1

INTRODUCTION

1.1 Machine Tool Multi-Speed Gearbox

In all metal cutting machine tools it is necessary to provide a range of rotational speeds at the spindle where the material to be cut is held. Using mechanical drives in the form of multi-speed gearboxes, the range covering several speeds is obtained from a single source of power which is in the form of an electric motor. It is necessary to get several different speeds at the spindle, because in any given situation of metal cutting, the required speed depends upon the type of material of the workpiece as well as of the tool, the machining accuracy required, the size of the work piece, the rate of material removal and the life of the tool.

A multi-speed gearbox located between the driving motor on one end and the spindle on the other, allows the operator to select a particular speed and obtain desired machining conditions.

In general the electric motor is first connected to a single stage or multi-stage belt drive and the output

from the belt drive is taken in as the input for gearbox.

Design of multi-speed transmission gearboxes for machine tool applications has been an important activity of designers.

A detailed theoretical exposition of how several design considerations are taken into account has been given by [1,2]*.

The overall process of the design of a machine tool gearbox can be divided into two stages.

In the first stage the designer develops a kinematic scheme showing, how different speeds can be obtained at the spindle shaft from one single speed of the motor. This is accomplished by employing a series of intermediate shafts and a number of gear pairs between successive pairs of intermediate shafts. Given the minimum speed required at the spindle, the maximum speed required at the spindle and the total number of speeds required at the spindle, it has been recommended that the intermediate speeds should be such that the total range of speeds form a series in Geometric Progression [1].

The second stage consists of designing all the constituent gears of the gearbox. This involves not only

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* Numbers in bracket refer to the References.

designing the gears on the basis of strength and wear considerations, but also involves generating the necessary data required for the production and inspection procedure of the gears. Detailed procedures for gear designs have been given by [3,4,5].

The selection of the number of intermediate shafts, the number of gear pairs in successive pairs of these shafts and the transmission ratio of each gear pair along with the input speed of the motor and the output speeds at the spindle determine broadly the kinematic scheme of transmission.

Design of kinematic scheme of gearbox has been attempted by [6,7].

Mathematically selection of a kinematic scheme or structure is a problem of type synthesis rather than of number synthesis. Moreover the number of alternative kinematic structures is a finite set. Because of the complexity of requirements of space and speed considerations, the problem of kinematic structure synthesis is difficult to model in a mathematical form.

As regards the second stage of design is concerned, several research workers have developed schemes for optimal design of gearboxes [8,9,10,11]. Majority of these works are based on mathematical programming techniques. In all these cases, an optimal design based on the criteria of

minimum weight or size has been sought, subjected to the constraints of stresses and deflections. The design parameters are the modules, the widths, spacing between gears etc. The solution procedure employed is generally the penalty function approach of non-linear programming [12,13].

1.2 Computer Aided Interactive Graphical Design

Engineering design is a science, an art and a practice, all rolled into one. Traditionally engineers had been relying on design charts, nomograms, thumb rules and experience for taking design decisions. Even then the designers used to take the help of such computational aids as the logarithmic tables, the slide-rules and the pocket calculators. With the ever increasing complexity of engineering systems, it has been found that such aids as the nomograms and the slide-rules are inadequate to cope with the volume of analysis and synthesis procedures.

Moreover engineering design is a process involving several iterations. An engineering system is generally conceived with certain assumptions and keeping in mind the specifications required. The process of conception is called synthesis. After synthesizing, the system is analysed to see whether it will perform satisfactorily with a certain degree of reliability and factor of safety. In case it does not pass the tests of analysis, the conceptualized model

is either modified or changed and the entire process is repeated until satisfactory results are obtained.

In short, the complex nature of engineering systems and the lengthy iterations of synthesis-analysis have made the tasks of designer difficult. Simultaneously the advent of computers have shown that interactive computing methods can substantially alleviate the above mentioned difficulties of the designer.

Five major engineering design functions can be wholly or partially turned over to the computer; design logic, equations and computations, design checking, engineering paperwork generation and graphical layout of design [14].

In design logic the computer's decision making capability can be utilized in selecting the best of a number of possible design alternatives. Of course, the creative effort would be put forth by the designer, and the computer would then be told via its program what to look for in picking a best design.

Almost all the equations and computations either are or can be put in a form suitable for computer solution. In the analytical stages many designs require numerous iterative calculations, most of which can be easily programmed for rapid computer solution.

Design checking can be a very time consuming manual task, especially if numerous critical calculations are involved. A separate computer program can be written to perform this basic function, with appropriate data substituted for specific design checks. Such a program could have error stops built into it, so that the computer would stop whenever an error was detected and automatically indicate the location of the error by giving a massage.

The computer is ideally suited for use in generating much of the engineering paperwork necessary to the successful implementation of a design.

The last and the most fascinating aspect is to generate the graphical layout of design and to get a hard copy record of that. To produce images whose appearance and motion make them quite unlike any other form of computer output, is an extremely effective medium for communication between man and computer. The human eye can absorb the information content of a displayed diagram much faster than it can scan a table of numbers.

The term interactive design means, the observer has some control over the intermediate results as well as final results, while the program is in execution mode. Designer is provided with the facility of selecting the required options, and/or sometimes allowed to change the

values of parameters calculated. All this logic is so developed that the program will automatically take care of the changes to be made according to the new information given by user. Thus interactiveness of the program is a very powerful tool, especially for mechanical designers. The main advantage of interactive programming is, the speed with which the user can visualize the effect made by changes. With the ability to interact with the computer, the designer can quickly correct a design error and can see a revised picture on the screen.

1.3 Problem Statement

In the present work, a Computer-Aided interactive, graphical method of designing multi-speed gearbox for a General Purpose Machine Tool has been developed. The input drive is from an A.C. motor of constant speed. The mode of gear changing is assumed to be of sliding type. Total transmission is broken up into various stages and gear pairs. Input drive is from the first shaft and output speeds are taken from the last shaft. The design procedure includes not only the synthesis of the kinematic structure, but also the design of all gears along with their production and inspection data.

The overall design process is considered to be divided into following stages:

1) Design of the kinematic structure of the gearbox.
2) Selection of module, pressure angle, helix angle (if any), appropriate number of teeth on constituent gears and distance between various stages, that is Geometric Calculations.
3) Stress analysis or strength consideration. It includes the determination of proper face width based on the bending strength and surface wear considerations, also calculation of pitch line velocity of gears is taken up.
4) Generation of gear correction data and also data for the inspection of individual gears.
5) Graphical display of all the results obtained and their proper tabulation.

Each one of the above design stages are described in subsequent chapters.

1.4 Scope of the Present Work

It has been felt that in designing a gearbox for General Purpose Machine Tools, a non-linear optimization approach is not always desirable from practical point of view. Moreover the schemes are computationally very expensive. Besides it is not possible to consider the aspect of production and inspection data in optimization schemes. These programs are not suitable for interactive

designing. Quite often designer would like to deviate some of the parameters from optimum depending upon the geometric or side constraints. Also sometimes designer would like to impose his own decisions over and above the calculated ones.

Keeping in view the above-mentioned features, a package is developed, which incorporates the various steps actually followed in day-to-day design of gearboxes in industries. The algorithm takes in consideration the various features of Machine Tool gearboxes.

The basic idea behind this work is to develop a compact package, which is not problem dependent. This program will need some input data to start the computation, but after that it will take its own decisions and calculate the various design parameters. Apart from it, at every stage it interacts with the designer (giving an option to make changes, to jump some unwanted portion of the program or sometimes gives warning if some input data is not matching with the expected one), and proceeds further only after getting signals from user.

After all the calculation part is over, the results can be displayed in the form of corresponding speed diagram, line diagram of gearbox and pitch line velocity distribution in various stages.

Now one can again go up in the program to make some changes and then see the corresponding effects in the form of modified diagrams. Important correction data for the gear teeth has also been generated in the same program. There are many iterative loops with the program which selects the best outcome of all the iterations.

The various parameters and inspection data calculated and needed for actual implementation of design are programmed to come in well documented tabular form, which is a must for keeping a record.

To prove the effectiveness of the program, three examples are taken up. A sample output of the tabular form of results, graphical outputs and the type of questions asked during the execution of the program are also given.

CHAPTER-2

GEARBOX DESIGN-I (KINEMATIC CONSIDERATIONS)

2.1 Introduction

The method generally adopted for the kinematic calculations of machine tool multi-speed gearboxes can be considered as graphical. It involves construction of several structural diagrams and speed charts.

Often the speed requirements on the spindle cannot be fulfilled by a direct transmission from the input shaft to the output shaft because of the limitations like -

(i) There may be many gear pairs on one shaft, causing too long shafting and hence loss of rigidity.

(ii) Transmission ratios may become too large, which will result in too large variations in pinion and gear sizes and further in rapid failure of gears.

Hence the spindle speeds are obtained by suitably breaking the total transmission into groups each group having different branching in it. Figure 2.1 represents the gearing diagram of a multi-speed gearbox.

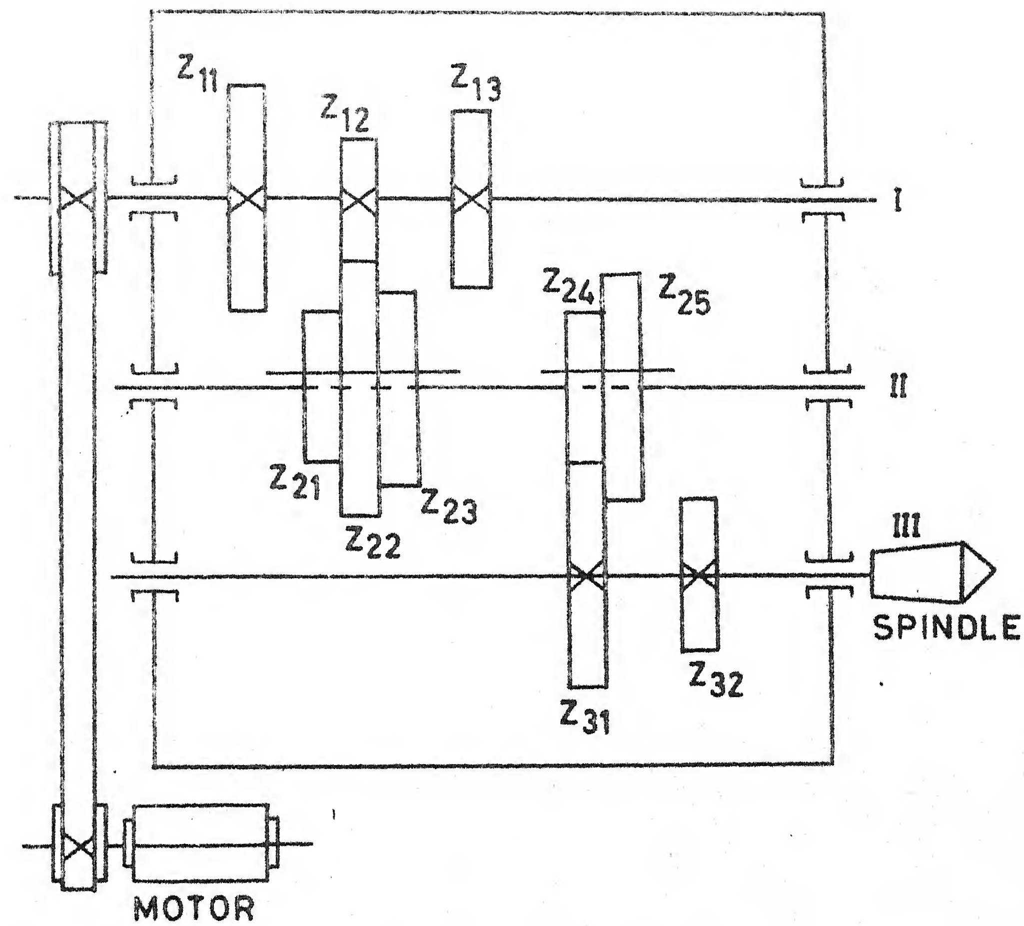


FIG.2.1 KINEMATIC DIAGRAM OF 6-SPEED GEAR BOX

2.2 Structure Diagram

It is the graphical representation of drive arrangement or the tree structure of gearbox, in its most general form, without considering the actual speed values and transmission ratios. Thus for the same input and output speeds, there are usually more than one structural diagrams. The number of structural diagrams is limited by some practical considerations [15]. Difference in the various structural diagrams lies in forming different number of groups, different branching within groups and in the ratio of two consecutive transmissions from a shaft. (Figure 2.2)

In obtaining different speeds by consecutive engagements of transmission within each group, the number of spindle speeds is equal to the product of the transmission numbers in each consecutive group [16].

$$Z = p_a \, p_b \, p_c \, \ldots\ldots\ldots p_{N_s} \qquad (2.1)$$

where

Z - number of spindle speeds

p_i - number of transmissions from the ith.stage

N_s - number of groups.

Thus with the number of spindle speeds given, the number of groups, the number of transmissions within each group and the group arrangement may be different. It is

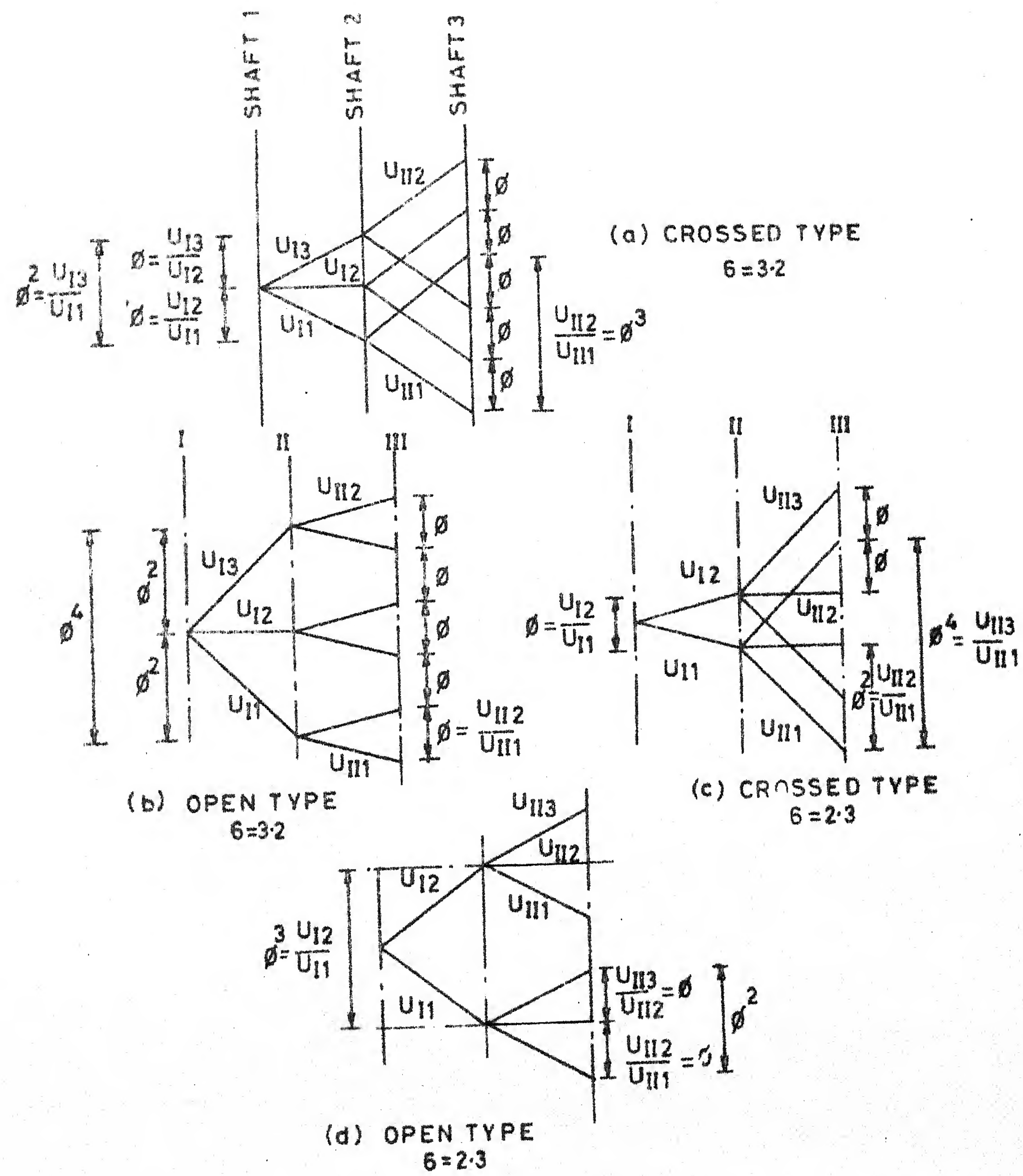


FIG. 2·2 STRUCTURAL DIAGRAMS OF 6-SPEED GEAR BOX

mainly the choice of these variables that determines the kinematic structure and the layout of the gearbox.

For the most common values of Z, the following variants may be used:

Z = 4 = 1[2] . 2[2]

Z = 6 = 1[2] . 2[3] = 1[3] . 2[2]

Z = 8 = 1[2] . 2[2] . 3[2]

Z = 12 = 1[3] . 2[2] . 3[2] = 1[2].2[3].3[2]=1[2].2[2].3[3]

Z = 16 = 1[2] . 2[2] . 3[2] . 4[2]

Z = 18 = 1[2] . 2[3] . 3[3] = 1[3].2[2].3[3]=1[3].2[3].3[2]

Z = 24 = 1[3] . 2[2] . 3[2] . 4[2]=1[2].2[3].3[2].4[2]

= 1[2] . 2[2] . 3[3] . 4[2]=1[2].2[2].3[2].4[3]

In the present discussion it is assumed that maximum number of transmissions from a speed in any stage is limited to three [15].

Thus the structural diagram provides the following data on the drive:

1. The number of transmission groups (a transmission group is a set of gear trains arranged on two consecutive shafts).

2. The number of transmissions in each group.

3. Relative order of the groups in the transmission train.

4. Group characteristics and relation between the

transmission ratios and speed range ratio of each group and of whole transmission (Figure 2.2).

5. Number of speed steps of each shaft in each group.

Before drawing a structure diagram which will provide the required spindle speeds, the information needed is :-

(i) Speed range, B ,

(ii) Number of steps, Z,

(iii) Ratio between steps, ϕ.

Following relationship exists between these three parameters

$$B = \frac{(SP)_{max}}{(SP)_{min}} \tag{2.2a}$$

$$(SP)_{max} = \phi^{(z-1)} \cdot (SP)_{min} \tag{2.2b}$$

$$\phi^{(z-1)} = \frac{(SP)_{max}}{(SP)_{min}} = B \tag{2.2c}$$

where

$(SP)_{max}$ - Maximum spindle speed

$(SP)_{min}$ - Minimum spindle speed.

In case of multi-stage reduction, if spindle speeds are in Geometric Progression with a common ratio ϕ , the ratio of any two consecutive speeds from any intermediate shaft becomes some integer power of the ratio ϕ. This

power index (say x) is called the group characteristic.

Two fundamental forms of structure diagrams are Open and Crossed. When the paths (that is connection between the input and the output points) do not cross each other, the distribution pattern is Open. If the paths cross each other the pattern is called Crossed. (Figure 2.2). Further there may be combination of above two forms. Also sometimes there may be discontinuities (to be discussed in the following paragraph) in speed variation on certain intermediate shafts. (Figure 2.3). Standard structure diagrams for various numbers of spindle speeds and their feasibility is given in literature [1,15].

The relationship between the spindle speed ratio and the speed ratios for other intermediate shafts for some different cases is shown in Figure 2.2.

A new term, namely, discontinuity in speed variation on any intermediate shaft is used in the present work to uniquely define non-standard form of structure diagram. This specification facilitates inputing the data. To understand this term, see the difference in diagrams of Figure 2.3.

Structure diagrams 2.3a and 2.3b are of type 2 x 3 x 2 having open-open-closed structure in the first, second and the third stages respectively. Similarly 2.3c

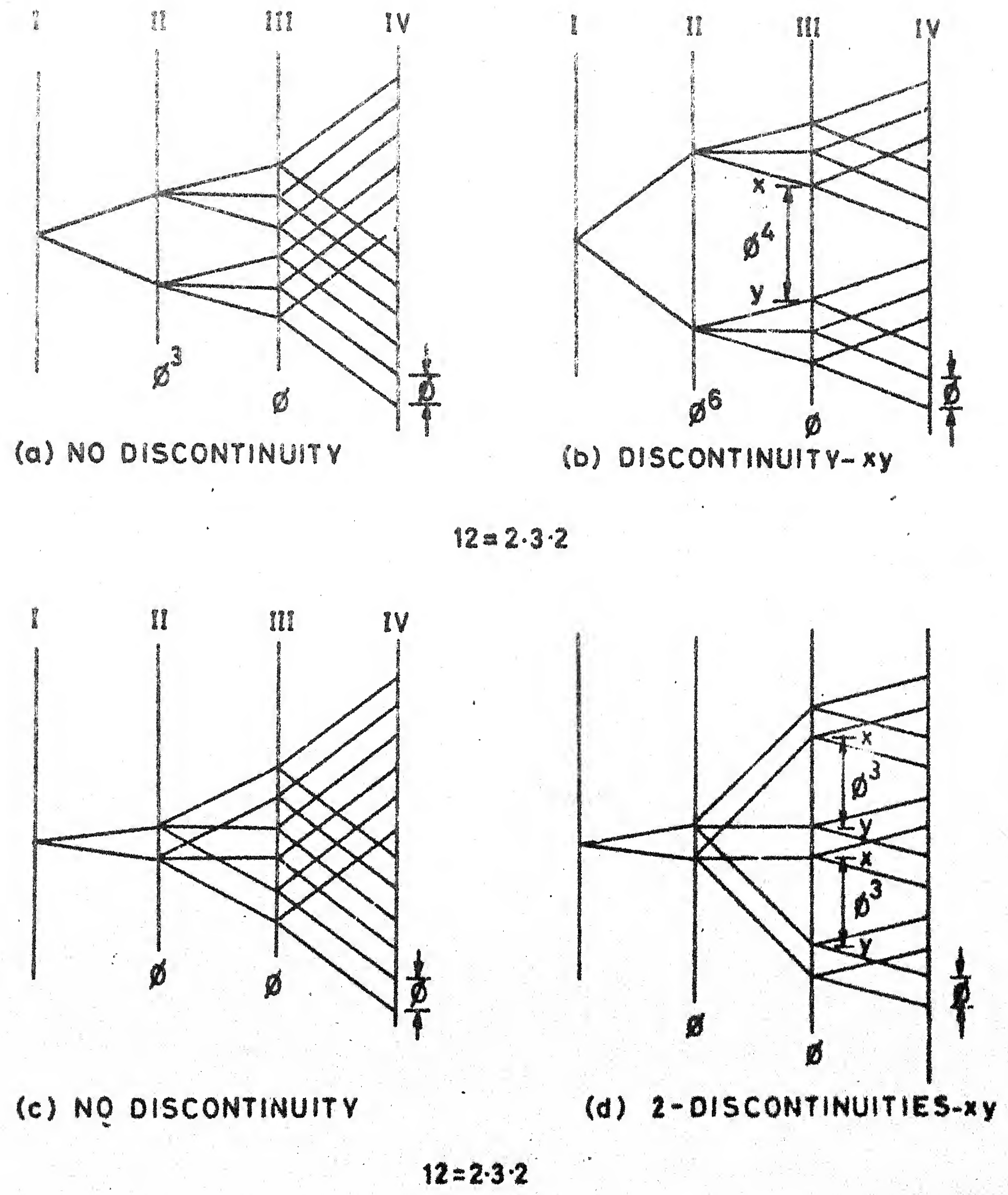


FIG. 2·3 STRUCTURAL DIAGRAMS OF 12 SPEED GEAR BOX WITH AND WITHOUT DISCONTINUITIES

and 2.3d are of open-closed-closed type structure in respective stages. But in diagrams 2.3b and 2.3d there is an intentional gap among speed distribution of intermediate shaft III. This gap namely XY happens to be some integer power of spindle speed ratio $\emptyset$. There can be one, two or more discontinuities depending upon the type of structure chosen.

Thus by including this feature in the programme, it will be possible to analyse any possible non-standard structure diagram and further to convert that into corresponding speed diagram.

2.3 Speed Diagram

Having devided the type of structure diagram, the next phase is to draw the speed diagrams and to select a suitable one.

Speed diagram serves to determine the specific values of all the transmission ratios and the speeds of all the shafts in the drive. It is constructed according to the structure adopted earlier.

For the same layout diagram, there is an infinite number of choices for the speed diagram, each providing a different design. In these diagrams logarithms of the actual speed values are plotted along the axis of the shaft. However, the points indicating the actual speeds

of different shafts are not arranged symmetrically, thus the lines joining the speeds between two axes are no longer arranged symmetrically as in the structure diagram. Though the equal transmission ratios are still represented by parallel lines. (Figure 2.4).

To draw speed charts, the program needs following input data:-

1) Speed of input motor (SP_m).
2) Number of stages (N_s).
3) Number of branches in each stage (p_i, $i = 1, \ldots . N_s$).
4) Spindle speed ratio ($\emptyset$).
5) Speed or group characteristics of intermediate shafts (x_i, $i = 2, \ldots . N_s$).

x_i's have to be integers, such that the speed ratio (R) of the i^{th} shaft is given by

$$R_i = \emptyset^{x_i} \qquad i = 2, \ldots . N_s \qquad (2.3)$$

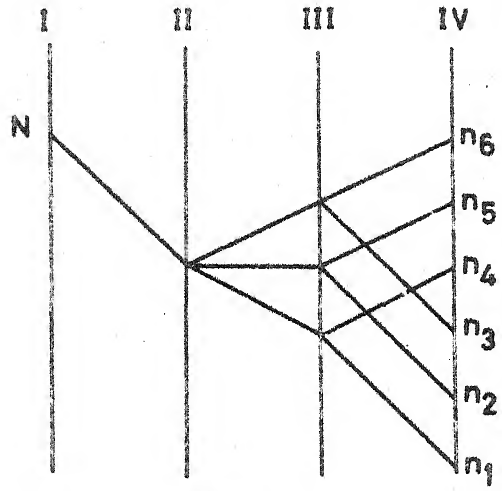
6) Order of discontinuity (if any) on intermediate shafts, as some integer power of $\emptyset$. Also specify the number of discontinuities on the shaft. It is presumed that the magnitude of all the discontinuities on a shaft be equal.

Let

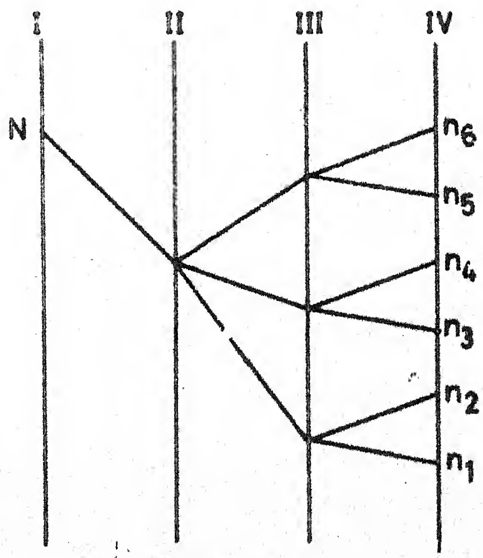
y_i - order of discontinuity on the i^{th} shaft,

C_i - number of discontinuities on the i^{th} shaft,

S_i - magnitude of discontinuity on the i^{th} shaft.



1x3x2
OPEN-CROSSED
(a)



1x3x2
FULLY OPEN
(b)

FIG. 2·4 SPEED DIAGRAMS OF 6 SPEED GEAR BOX

Then S_i in terms of spindle speed ratio is given by

$$S_i = \phi^{y_i} \qquad (2.4)$$

$$i = 3, \ldots .(N_h - 1)$$

where

N_h - represents total number of shafts

$= N_s + 1$

According to present definition discontinuities can not occur on the first, second and the last shaft.

7) Lowest speed on every shaft, except the first shaft, whose speed is the speed of motor.

Let it be represented by SL_i , $i = 2, \ldots .N_h$

8) Maximum value of transmission ratio permissible in any stage (U_1).

After getting these inputs, the programme will start calculating all the speeds and transmission ratios according to the following formulation.

2.4 Calculation of Number of Speeds on Successive Shafts

Let NS_i represents total number of speeds on the i^{th} shaft.

First shaft is always assumed to have only one speed, and that is the speed of the input motor.

If N_h represents the total number of shafts, then we have,

$$N_h = N_s + 1 \qquad (2.5)$$

where N_s is the total number of stages.

Further the number of speeds can be obtained for the successive shafts using following relations

$$NS_i = NS_{i-1} \cdot p_{i-1} \qquad (2.6)$$

$$i = 2, \ldots . N_h$$

and $NS_1 = 1$

where p_{i-1} is the number of branches in the stage (i-1).

2.4.1 Calculation of Actual Speeds on Shafts

All the speeds of a shaft are calculated on the basis of the given lowest speeds on every shaft and the speed ratios.

If SP_{ij} is the j^{th} speed of the i^{th} shaft, it is given by the following recurrence relation:

$$SP_{ij} = SL_i \cdot \phi^{[x_i \cdot (j-1)]}$$

$$j = 1, \ldots . NS_i \qquad (2.7)$$

$$SP_{11} = SP_m$$

where

SL_i - lowest speed of the i^{th} shaft

x_i - speed characteristic of the i^{th} shaft

ϕ - spindle speed ratio

SP_m - speed of input motor

It is to be noted that Equations (2.7) are valid for those shafts on which discontinuities are not present.

If discontinuities are there on a shaft then above relation is slightly modified. According to diagrams in Figure 2.3, maximum number of discontinuities may be equal to one less than the maximum number of branching allowed from a source point in any stage. Thus in the present work, there may be one or two discontinuities on a shaft.

Let the number of discontinuities on the i^{th} shaft be C_i, each of magnitude S_i, given by

$$S_i = \phi^{y_i} \tag{2.8}$$

where

y_i - order of discontinuity.

Then the total number of speeds on i^{th} shaft can be divided into $(C_i + 1)$ different clusters of speeds, each separated from another by one discontinuity. Figure 2.3. At the most there can be three different clusters of speeds on a shaft.

For the first cluster, which lies before the first discontinuity, the speeds are calculated as by the Equations 2.7.

Speeds just after the first discontinuity are given by the relation

$$SP_{ij} = SL_i \cdot \phi^{[x_i \cdot (j - 2) + y_i]} \tag{2.9}$$

Speeds after second discontinuity are calculated from the recurrence

$$SP_{ij} = SL_i \cdot \phi^{[x_i \cdot (j - 3) + 2y_i]} \tag{2.10}$$

2.5.2 Calculation of Transmission Ratios

Let NT_i represents the number of transmission ratios in the i^{th} stage.

Then

$$NT_i = p_i \qquad i = 1,\ldots N_s \tag{2.11}$$

where

p_i - number of branches in the i^{th} stage.

It is assumed in the present work, that the maximum number of transmissions from a point in any stage is less than or equal to 3.

Let U_{ij} be the j^{th} transmission ratio in the i^{th} stage, and is defined by the ratios

$$U_{ij} = \frac{\text{Speed on } (i + 1)^{th} \text{ shaft}}{\text{Speed on } i^{th} \text{ shaft}} \tag{2.12}$$

and also,

$$= \frac{\text{Number of teeth on driver gear}}{\text{Number of teeth on driven gear}} \tag{2.13}$$

Depending upon the type of structure within a stage, two different cases arises for the calculation of transmission ratios.

Case-I Open type structure. (Figure 2.5a).

In this case the speed step ratios for the i^{th} and the $(i+1)^{th}$ shaft are different.

$$R_i \neq R_{i+1}$$

Considering the case of three transmissions, the transmission ratios are given by the following recurrence formula

$$U_{ij} = \frac{SP_{i+1,j}}{SP_{i,1}} \qquad j = 1,2,3 \tag{2.14}$$

where

$SP_{i,1}$ represents first speed of the i^{th} (input) shaft

$SP_{i+1,j}$ represents j^{th} speed of the $(i+1)^{th}$ (output) shaft.

In this case the presence of discontinuities do not effect the above formulation.

Case-II Close type structure. Figure (2.5b and c).

Here the speed step ratios for the i^{th} and the $(i+1)^{th}$ shaft are same

$$R_i = R_{i+1}$$

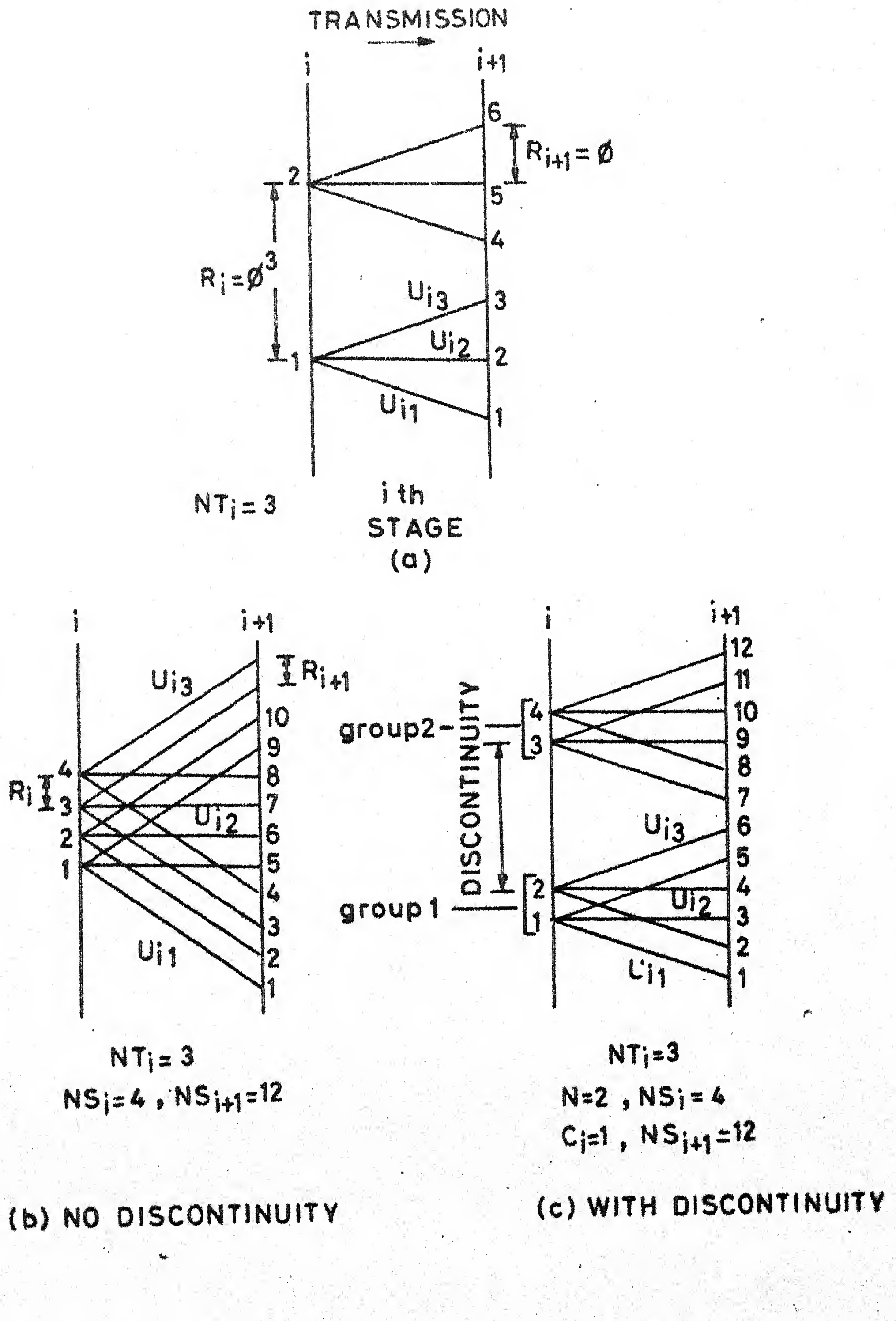


FIG. 2·5 STRUCTURAL DIAGRAMS TO SHOW THE CALCULATION OF TRANSMISSION RATIOS

If there are one or two transmissions from a source point, then the discontinuities do not effect the formulation, but in case of three transmissions the formulation differs from that of no discontinuity situation.

Thus,

(i) if $NT_i = 1$, then

$$U_{i1} = \frac{SL_{i+1}}{SL_i} \tag{2.15}$$

where

SL_i - lowest speed on the i^{th} shaft. (Input shaft for i^{th} stage).

SL_{i+1} - lowest speed on the $(i+1)^{th}$ shaft. (Output shaft for i^{th}. stage.)

(ii) if $NT_i = 2$, then

$$U_{i1} = \frac{SL_{i+1}}{SL_i}$$

$$U_{i2} = \frac{SP_{i+1,\ NS_{i+1}}}{SP_{i,\ NS_i}} \tag{2.16}$$

where

SP_{i,NS_i} - last speed of i^{th} shaft

SP_{i+1,NS_i} - last speed of $(i+1)^{th}$ shaft.

(iii) if $NT_i = 3$, then

$$U_{i1} = \frac{SL_{i+1}}{SL_i}$$

$$U_{i3} = \frac{SP_{i+1,\ NS_{i+1}}}{SP_{i,\ NS_i}} \qquad (2.17a)$$

Calculation of U_{i2} depends upon whether the discontinuities are present or not.

(a) No discontinuity. (Figure 2.5b)

$$U_{i2} = \frac{SP_{i+1,\ (NS_i+1)}}{SP_{i,1}} \qquad (2.17b)$$

(b) With discontinuity. (Figure 2.5c).

Here the total number of speeds on the i^{th} shaft can be divided into separate groups (or clusters of speeds) each having 'N' number of speeds given by

$$N = \frac{NS_i}{(C_i+1)} \qquad (2.17c)$$

where

C_i - number of discontinuities on the i^{th} shaft.

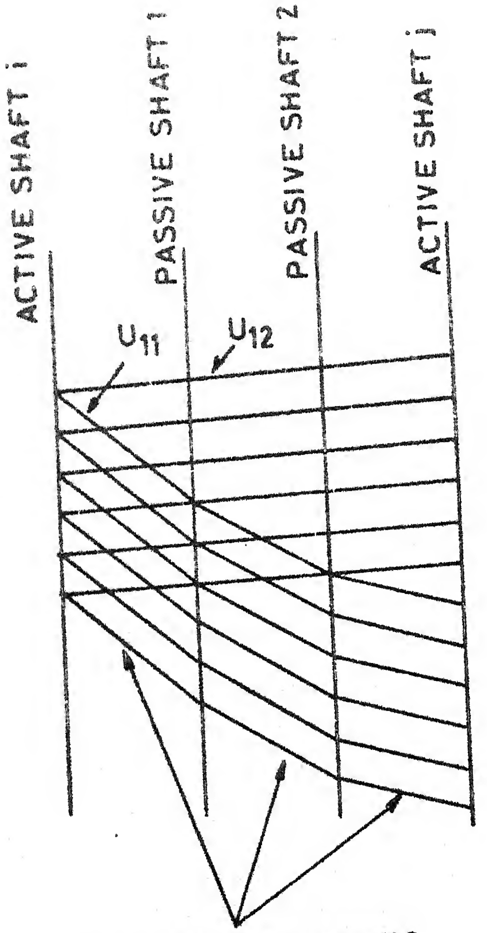
Then the second transmission ratio is given by

$$U_{i2} = \frac{SP_{i+1,\ N+1}}{SP_{i,1}} \qquad (2.17d)$$

2.6 Inclusion of Passive Stages, if Necessary

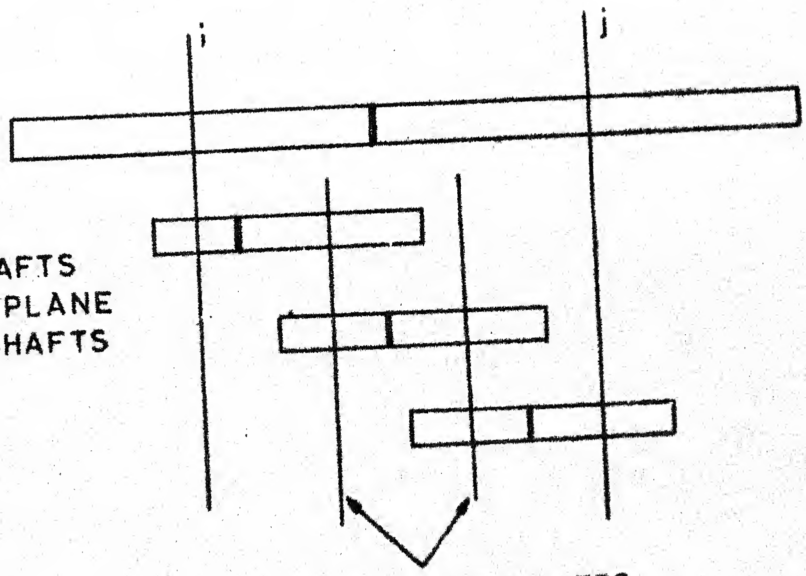
Above calculated transmission ratios are based on the lowest speeds specified on each shaft. Sometimes it may happen that, one of the transmission ratio in some stage exceeds the maximum value specified in the design, though all other transmissions of that stage are quite acceptable. In such case, instead of increasing one whole stage, it is beneficial to reduce only that particular transmission through one or more sub-stages by inserting the additional shafts in-between. (Figure 2.6).

To differenciate between the main shafts and the additional shafts inserted because of the above reason, the main shafts are named as Active shafts and the additional shafts as Passive shafts. These names signify that the active shafts are those shafts which will be active (that is running) all the time for any spindle speed, on the other hand the passive shafts are active only when those particular speeds are desired which can be obtained, only by engaging that transmission line which was earlier exceeding, and lowered down within prescribed limits by inserting these shafts. The additional sub-stages so formed within that stage consists of only one gear pair and are called as the sub-stages or the passive stages.



TRANSMISSION RATIO U_{11} IS REDUCED IN THREE SUB-STAGES, THOUGH THE SECOND TRANSMISSION IS DIRECT FROM SHAFT i TO j

(a) PASSIVE STAGES



PASSIVE SHAFTS ARE NOT IN PLANE OF ACTIVE SHAFTS

(b) PASSIVE SHAFTS

FIG.2.6 PASSIVE TRANSMISSION LINE

Present program can take care of the above situation through two different channels.

In the first case inclusion of passive shafts is done by the following stepwise procedure.

Step-1 Keep first sub-stage transmission ratio as the largest permissible, that is U_1, which is prespecified.

Step-2 Calculate the speeds thus obtained on first passive shaft.

Step-3 Calculate the remaining transmission ratio to obtain the required spindle speeds from the speeds of the passive shaft. Check, if it is within the limit, then stop the insertion of any more passive shaft, otherwise repeat the above steps for second passive stage.

Another way of taking care of passive stages and such transmissions, is to give freedom to the designer to fix the number of passive reductions and to fix the corresponding transmission ratios. However the inputs should be well within the prescribed limit of transmission ratio.

Example of a 24-speed gearbox given in Chapter-5 exhibits both the above schemes.

CHAPTER-3

GEARBOX DESIGN-II (GEOMETRIC CONSIDERATIONS)

3.1 Introduction

After deciding about the transmission ratios and the various speeds in successive stages, the next decision is about the geometry of gears and the center distances between them, so as to obtain the required spindle speeds.

Input data required for this phase of design are as follows:

1. Values of normal modules for each gear pair in mm.
   m_{ij} represents the module of j^{th} gear pair of i^{th} stage.
2. Helix angle, specifying the gear pair and the stage.
   β_{ij} represents the helix angle on pitch circle, in degrees, of j^{th} gear pair in i^{th} stage.

   This needs to be specified only if the value of β is other than zero.
3. Pressure angle on pitch circle. (Normal section in case of Helical gears).
   α_{on} represents the pressure angle and is assumed to be constant throughout.

4. Horse power of the motor in KW., represented by P.

5. Minimum number of teeth permissible, ZMIN.

The assumption that the parameters like ; modules, pressure angle, helix angles are to be supplied by the designer, will provide flexibility to make some adjustments in the design, by changing one or some of these parameters at any time. Here the modules can take either same value throughout or can have different values for different gear pairs. Helix angle specification is optional default value is automatically assumed to be zero. In case of the helical gears, modules specified are assumed to be the normal modules. All the formulas used are such, that with zero helix angle they reduce to spur gear relations.

A minimum value of the number of teeth permissible is to be specified by the designer. Further the minimum number of teeth in each stage will depend upon the ratio of, relative magnitude of the maximum transmission ratio permissible to the maximum transmission ratio of that stage. If the transmission ratio in some stage happens to be the maximum permissible, then the minimum number of teeth in that stage will be equal to the prescribed minimum value, otherwise it will always be somewhat higher than the limiting value. Thus in the present work, magnitude of the maximum transmission ratio is the deciding

factor for fixing the minimum number of teeth in a particular stage.

In order that the minimum number of teeth in any stage should not vary by a large amount, presently the factor used to multiply prescribed minimum is square root of the above mentioned ratio. Though there is no hard and fast rule for it, one can have also other relations for the above ratio.

One of the important feature of the program at this stage is, that it is not mandatory for the designer to accept the minimum number of teeth calculated in each stage by the algorithm. A question will be asked, (displaying the calculated value, corresponding stage and the transmission ratio) whether the designer wants to change it or not. If some other value is assigned as the answer of above question, then that will be taken up as the minimum number of teeth for that particular stage. This facilitates the designer in reducing the size of gears and hence the overall size of the gearbox, subject to fulfilling the strength requirements satisfactorily.

After the number of teeth and the center distance calculation for a stage is over, the actual values of transmission ratios and various speeds of the driven shaft actually obtained are calculated, based on the number of

teeth. The variation in the values of above quantities is, because of converting the number of teeth calculated into suitable whole numbers, but this variation should be within 3 to 5%.

The mathematical formulation and step-by-step procedure for calculating the above-mentioned parameters are as follows:

3.2 Calculations for Fixing the Minimum Number of Teeth in Any Stage

To start the calculation, first of all, take the inverse of those transmission ratios which are less than 1, and then select the maximum from a set of original and/or inverted values of all the transmission ratios of the present stage. Gear pair corresponding to this maximum will require the least number of teeth, either on its driver gear or driven gear.

Let U_m represents the maximum mentioned above for the i^{th} stage and it corresponds to the k^{th} transmission ratio U_{ik}.

Let Z_{1ij} and Z_{2ij} represents the number of teeth on the j^{th} gear of the i^{th} stage on the driver and driven shafts respectively.

Depending upon whether transmission ratio is less than, or greater than, or equal to one following three

different cases arise.

Case-I $U_{ik} < 1$

Calculation of the number of teeth is to be done according to the following steps:

Step-1: For a given value of U_{ik}, the calculated values of the numbers of teeth on driver and driven gears are

$$Z1 = ZMIN \cdot \sqrt{\frac{U_1}{U_m}} \qquad (3.1)$$

$$Z2 = \frac{Z1}{U_{ik}} \qquad (3.2)$$

where ZMIN and U_1 are prespecified input parameters.

Note that the values above obtained are not integer values.

Step-2: Prepare a set of 9 pairs of integral values representing the designed values of numbers of teeth on gear pair as

(Z11, Z21) , (Z11, Z22),......(Z13, Z23)

where Z11 = trunc (Z1) ; Z12 = trunc (Z1) + 1 ;

Z13 = trunc (Z1) - 1

Z21 = trunc (Z2) ; Z22 = trunc (Z2) + 1 ; (3.3)

Z23 = trunc (Z2) - 1

Step-3: For each gear pair defined in step-2, calculate the actual transmission ratios by taking the ratio of the number of teeth on the driver to that on the driven gear.

Step-4: For each transmission ratio computed in step-3, find the variation from the ideal transmission ratio U_{ik}. Select that pair for which the variation is minimum. Assign them to the number of teeth on driver Z_{1ik} and number of teeth on driven gear Z_{2ik}. In this case Z_{1ik} represents the minimum number of teeth for the i^{th} stage.

At this point a choice is given to the designer, whether to accept above calculated value, or to assign some other value to Z_{1ik}. In case a specific value is supplied, the steps of calculations will be as follows:

Step-1: For a given value of U_{ik} and Z_{1ik}, the calculated value of number of teeth for driven gear is

$$Z2 = \frac{Z_{1ik}}{U_{ik}} \qquad (3.4)$$

Again the value of Z2 above is not an integer.

Step-2: Prepare a set of 3 pairs of integral values representing the designed values of number of teeth on gear pair as

$(Z_{1ik}, Z21), (Z_{1ik}, Z22), (Z_{1ik}, Z23)$

where

$$Z21 = \text{trunc}\ (Z2)\ ;\ Z22 = \text{trunc}\ (Z2) + 1$$

$$Z23 = \text{trunc}\ (Z2) - 1 \qquad \ldots. \qquad (3.5)$$

Step-3: and Step-4: Same as described earlier

Case-II When $U_{ik} > 1$

The only difference in calculations of this case is in step-1, all other steps are same as in Case-I. Here the number of teeth on driven gear will correspond to the minimum for this stage.

Step-1: Calculated values of the numbers of teeth on driven and driver gears are

$$Z2 = ZMIN \cdot \sqrt{\frac{U_1}{U_m}} \qquad (3.6)$$

$$Z1 = Z2 \cdot U_{ik} \qquad (3.7)$$

Case-III $U_{ik} = 1$

Here the calculated values of the numbers of teeth on driver and driven gears are same and are given by

$$Z1 = Z2 = ZMIN \cdot \sqrt{\frac{U_1}{U_m}} \qquad (3.8)$$

The designed values are given by

$$Z_{1ik} = \text{trunc } (Z1)$$
$$Z_{2ik} = \text{trunc } (Z2) \tag{3.9}$$

3.3 Fixing the Center Distance Between the i^{th} and j^{th} Shafts Constituting the i^{th} stage

Center distance between the constituent shafts of i^{th} stage can be calculated on the basis of the number of teeth on the gear pair corresponding to the maximum transmission ratio as calculated above. Let a_{ik} is the ideal Center Distance for the k^{th} gear pair in the i^{th} stage, and is given by

$$a_{ik} = \frac{(Z_{1ik} + Z_{2ik}) \cdot m_{ik}}{2 \cdot \text{Cos } (\beta_{ik})} \tag{3.10}$$

where

m_{ik} - normal module for the k^{th} gear pair of the i^{th} stage

β_{ik} - helix angle for the k^{th} gear pair of the i^{th} stage.

3.4 Calculation of Number of Teeth on Other Gear Pairs of the i^{th} Stage

In this section the gears are selected to obtain all other transmissions of i^{th} stage, except k^{th} transmission for which the selection has been done.

The design constraints are, to keep the center distances for other gear pairs as much close to the one

already fixed and at the same time selection should be such, that the variations in the transmission ratios are minimum.

Consider the case of n^{th} gear pair of the i^{th} stage. If ZSUM represents the sum of the numbers of teeth ondriver and driven gears, then the design procedure is given in the following steps:

Step-1: Based on the precalculated center distance of the i^{th} stage, ZSUM for the n^{th} gear pair is given by,

$$ZSUM = \frac{2 \cdot a_{ik} \cdot \cos(\beta_{in})}{m_{in}} \tag{3.11}$$

Step-2: To calculate the number of teeth on driver and driven gears, knowing ZSUM and transmission ratio, following relationships exist,

$$ZSUM = Z1 + Z2 \tag{3.12}$$

and

$$\frac{Z1}{Z2} = U_{in} \tag{3.13}$$

Equations 3.12 and 3.13 togather give

$$Z1 = \frac{ZSUM}{\left(1 + \frac{1}{U_{in}}\right)} \tag{3.14}$$

$$Z2 = ZSUM - Z1 \tag{3.15}$$

Above calculated values of Z1 and Z2 are non-integers.

Step-3: There are two different ways to select a suitable pair of numbers of teeth having integral values.

(i) Based on Center Distance accuracy. In this case the values of Z1 and Z2 are simply rounded off to the nearest integer values. Thus the designed values of the number of teeth on driver and driven gears are given by

$$Z_{1in} = \text{Round off}(Z1)$$
$$Z_{2in} = \text{Round off}(Z2) \qquad (3.16)$$

In this case, no doubt the center distances of all the gear pairs in a stage are very close to each other, but the variation in the actual speed values from the desired are comparatively large.

(ii) Based on Transmission Ratio accuracy.
In this case the procedure to obtain the values of Z_{1in} and Z_{2in} is same as described in Art 3.2. Here the variation in the centerdistances of adjacent gear pairs is slightly larger than in the previous case, which can be adjusted by using profile shifted gears, but at the same time this method gives very little variation in the desired values of speeds of rotation. In the present algorithm this method is used.

Though, the calculation of the number of teeth on all gear pairs is done taking same center

distance, but due to rounding off errors the centerdistance for different gear pairs of the same stage vary slightly. Hence it is essential to calculate all centerdistances within a stage from the relation.

$$a_{in} = \frac{(Z_{1in} + Z_{2in}) \cdot m_{in}}{2 \cdot \cos(\beta_{in})} \qquad (3.17)$$

In case the current stage has one passive or indirect transmission as described in Art 2.6, then the calculations for the passive stage are done separately, assuming them as stages having only one transmission line. Calculation procedure for them is exactly similar to that described in Arts. 3.2 and 3.3. It is assumed that the passive shafts are not in the same plane as that of the active shafts. In such case the passive stage center distances wouldn't effect the main stage center distance.

After completing passive transmission calculations, the remaining transmissions of that stage are taken, and once again all the calculations are done starting from selecting the maximum transmission ratio as in Arts 3.2-3.4.

3.5 Calculation of Modified Transmission Ratios and Speeds

Modified transmission ratios and speeds are those which are actually obtained after selecting the gear pairs.

Let U_{mij} represents the j^{th} actual or modified transmission ratio in the i^{th} stage. It is given by

$$U_{mij} = \frac{Z_{1ij}}{Z_{2ij}} \qquad (3.18)$$

$$j = 1, \ldots . p_i$$

Modified speeds of the i^{th} shaft are calculated by multiplying the modified speeds of the previous shaft by the corresponding actual transmission ratios.

Equations for these calculations depend upon the type of structure (open or closed) within the stage.

Case-I Fully open structure. Figure 2.5a.

Let SP_{mjk} represents the k^{th} modified or actual speed of the j^{th} shaft. It is given by the following recurrance relation.

$$SP_{m11} = SP_{11}$$

$$SP_{mjk} = SP_{min} \cdot U_{mil} \qquad (3.19)$$

$$n = 1, \ldots . NS_i$$

$$l = 1, \ldots . p_i$$

$$k = 1, \ldots . NS_j$$

where

SP_{11} - speed of the first shaft

SP_{min} - n^{th} modified speed of the i^{th} shaft

U_{mil} - l^{th} modified transmission ratio of the i^{th} stage.

Case-II Fully closed structure. Figure 2.5b

The only difference here is in the order , in which the subsequent multiplication is to be done. Here the recurrence relation is

$$SP_{m11} = SP_{11}$$
$$SP_{mjk} = SP_{min} \cdot U_{mil} \qquad (3.20)$$

$$l = 1, \ldots . p_i$$
$$n = 1, \ldots . NS_i$$
$$k = 1, \ldots . NS_j$$

Case-III Closed structure with discontinuities on input shaft. Figure 2.5c.

In such a case, recurrence relation to be used is same as in case (ii), except that the calculations will be done by dividing the speeds on the input shaft in groups and starting the recurrence relation afresh for each group.

As the modified speeds are different from the ideal speeds calculated earlier, therefore after the first stage calculations are over, modified values of the speeds

of the driven shaft, as calculated at the end of previous stage calculations, are used as the driver shaft speeds for all the calculations of the next stage. New transmission ratios are calculated from the modified speeds of the driver shaft and the ideal speeds of the driven shaft of that stage. These are different from the ideal transmission ratios calculated earlier. Also this is not to be confused with the modified or actual transmission ratio, as that will be calculated after fixing the number of teeth based on this new value of transmission ratio.

By this approach, the desired speed values of the successive shafts will be better obtained, because unlike the actual transmission ratios, now the new transmission ratios thus calculated are the ones which will give the desired driven shaft speeds from the modified speeds of driver shaft.

3.6 Calculation of Pitch Line Velocities

Pitch line velocity is the tangential velocity of gears at pitch point. In the given computer outputs, pitch line velocities are referred to the different stages. It includes the pitch line velocities of all the gear pairs constituting transmission line for that particular stage.

If the i^{th} stage between the i^{th} and the j^{th} shafts as driver and driven respectively, has 'n' number

of gear pairs in it, then

the total number of pitch line velocities in the i^{th} stage $= n \cdot NS_i$ (3.21)

where

$n = p_i$

NS_i = total number of speeds of the i^{th} shaft.

Corresponding values of pitch line velocities in the i^{th} stage are given by

$$V_{ir} = \frac{\pi \cdot d_{1ij} \cdot SP_{mik}}{60} \quad (3.22)$$

$j = 1, \ldots . p_i$

$k = 1, \ldots NS_i$

$r = 1, \ldots NP_i$

where

NP_i - number of pitch line velocities in the i^{th} stage

V_{ir} - r^{th} pitch line velocity within i^{th} stage in m/sec.

NP_i - number of pitch line velocities in the i^{th} stage

d_{1ij} - pitch circle diameter of the j^{th} driver gear of the i^{th} stage. (See Chapter-4).

3.7 Face Width Calculation

In the first calculation, face width of a gear pair is calculated by taking it within the limits 10m-12m, where 'm' is the module of gear pair. The above calculated value is then checked with the help of following formula, based on surface strength and also considering safety factors for pitting and shock.

$$Z^2 m^2 b \geqslant K \cdot \frac{(GR + 1)}{GR} \cdot \frac{P \cdot \eta}{(SP)_{low}} \qquad (3.23)$$

where

Z = No. of teeth

m = module in mm.

b = face width in mm

GR = gear ratio

$= Z_2/Z_1$

P = power of input motor in KW.

η = transmission efficiency

= 0.95 (assumed in present case)

$(SP)_{low}$ = lowest speed (r.p.m.) on which the particular gear may engage

K = constant

$= 2.85 \times 10^6$ for helical gears

$= 2.85 \times 1.2 \times 10^6$ for spur gears

But the above formula has limited applicability, because at very low speeds it gives large values for face width. In such cases one can reduce the factors of safety for that particular pair depending upon, how frequently that speed is to be used so as to reduce the face width. And also one may apply other tests to check it further.

In case 10m-12m face width does not satisfy the Equation 3.23, the value obtained from Equation is taken as final.

A provision is made to change these values by the designer.

CHAPTER-4

GEARBOX DESIGN-III

(GEAR CORRECTION AND INSPECTION DATA)

4.1 Profile-Shifted Involute Gearing

By means of profile shift, that is, by withdrawing the line of equal space and thickness for cutter 2 from the pitch circle 6 (Figure 4.1) by an amount Xm (where X is the profile shift for gear and m is module), it is possible to manufacture with the same tool gears with a larger average flank angle, a smaller number of teeth, and a larger load carrying capacity. Further it is possible to obtain exactly a specified center distance while sticking to a particular module.

Effect of possitive shift on gear tooth is, the base of tooth becomes wider and the crest width becomes smaller. The effect of profile shift on tooth form decreases with increasing number of teeth. The effect is nil when $Z = \infty$ (rack).

Profile shifted gear pairs have in operation a different pressure angle (α_b) and a different center distance 'a_c' compared to the standard gears (with α_o and a)

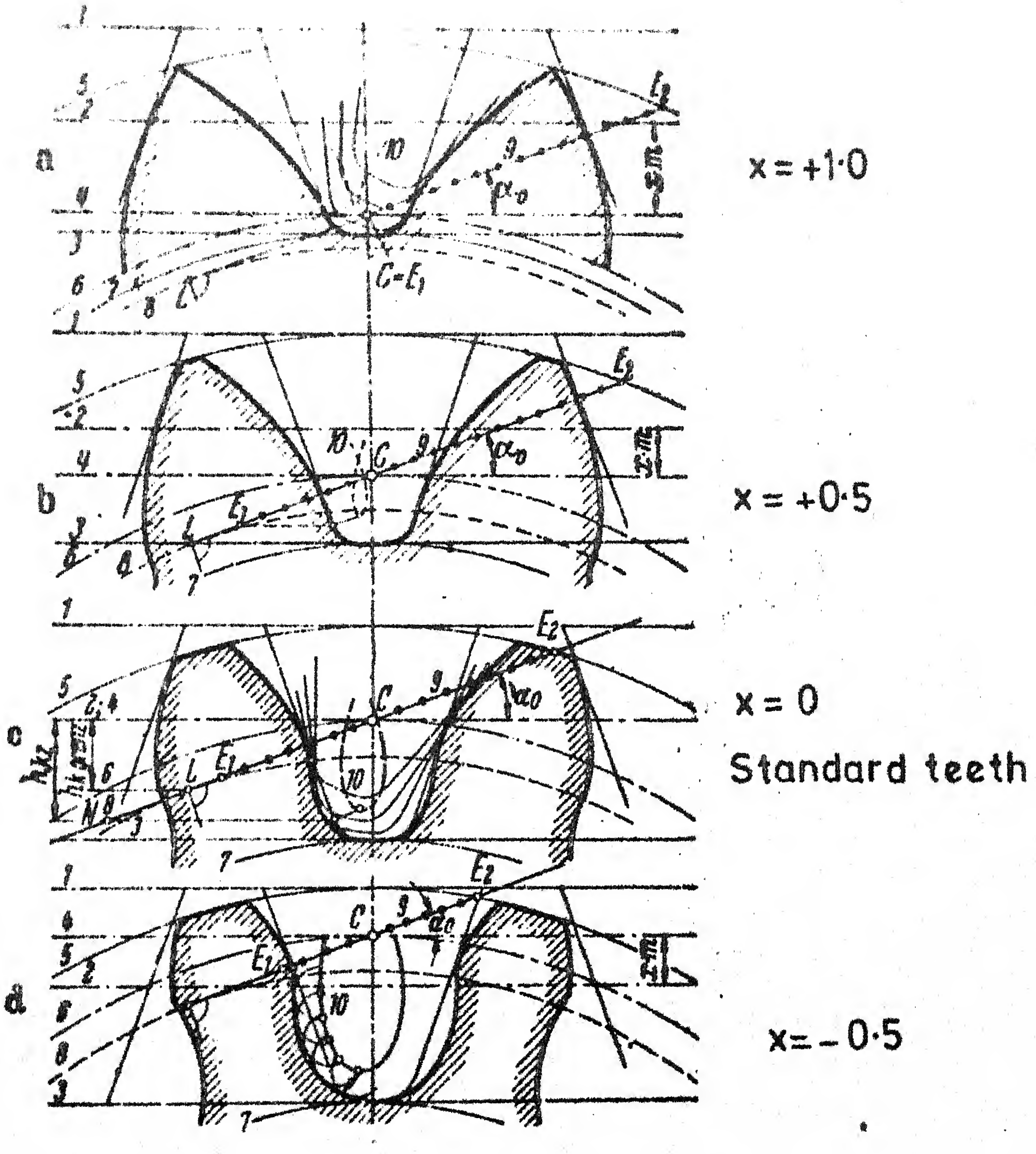


FIG. 4·1 EFFECT OF PROFILE SHIFTS ON A GEAR WITH 12 TEETH

2-LINE OF EQUAL THICKNESS AND SPACING FOR CUTTER ; 6-PITCH CIRCLE ; 8-BASE CIRCLE.

having the same numbers of teeth and generated by the same cutter. The operating pressure angle α_b increases with a_c/a. (Figure 4.2).

4.2 Calculation of Profile Shifts

The method for deciding the amount of profile shifts on each gear is based on the recommendations of DIN standard 3992. As per this standard, one computes first the sum of the profile shifts $m \cdot (X_1 + X_2)$ and then finds as to how the total shift is to be distributed on individual gears, mX_1 and mX_2.

The sum of the profile shift $m\ (X_1 + X_2)$ is computed from a chart shown in Figure 4.3. The value of total shift depends upon the operating conditions and the sum of the number of teeth on the pinion and the gear. The operating conditions are generally characterised by a set of zones and preferred line within each zone.

The distribution of the profile shifts on individual gears is computed using the chart given in Figure 4.4.

For a set of gear pairs belonging to a particular stage, the corrected center distance as well as profile shifts for all the gears are computed using the following procedure. Here the corresponding symbols are used without using the suffixes for stage and gear pair.

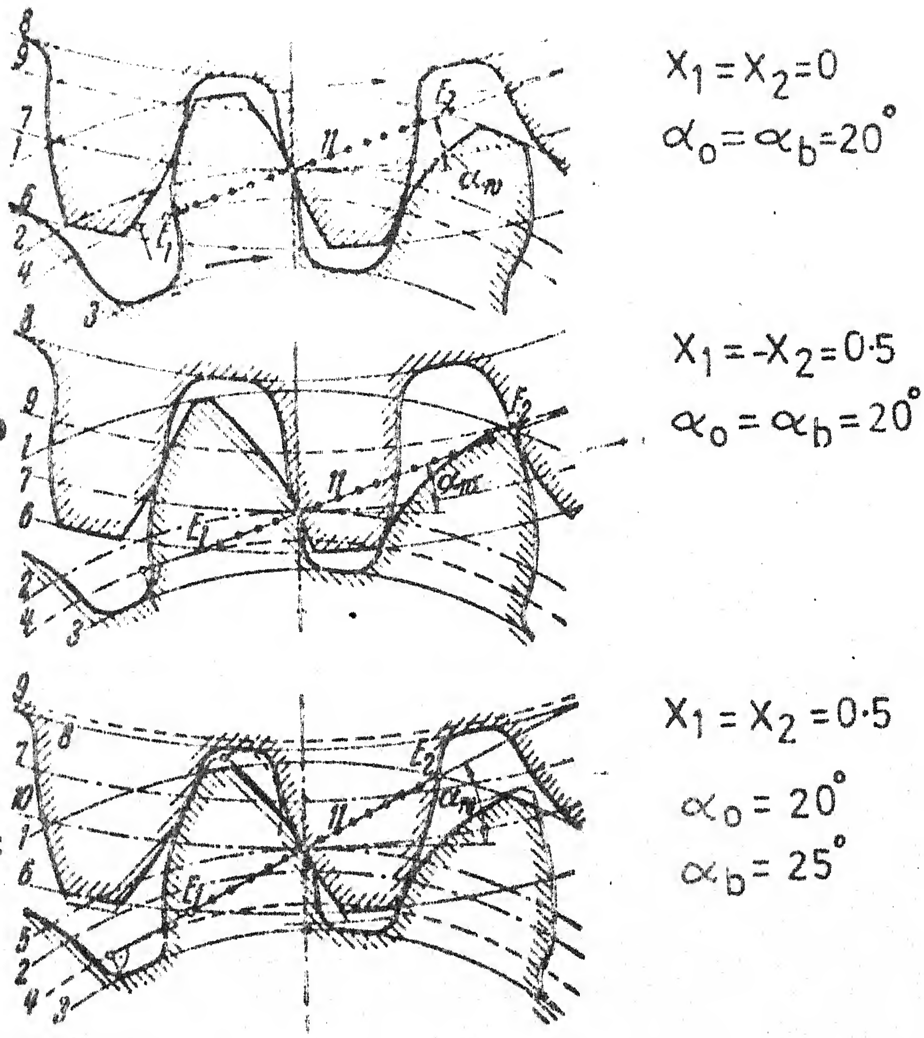


FIG. 4·2 GEAR PAIR WITH $Z_1=12$, $Z_2=25$ AT DIFFERENT PROFILE SHIFTS

1, 6 ADDENDUM CIRCLES

2, 7 PITCH CIRCLES

3, 8 DEDENDUM CIRCLES

4, 9 BASE CIRCLES

5, 10 ROLLING CIRCLES

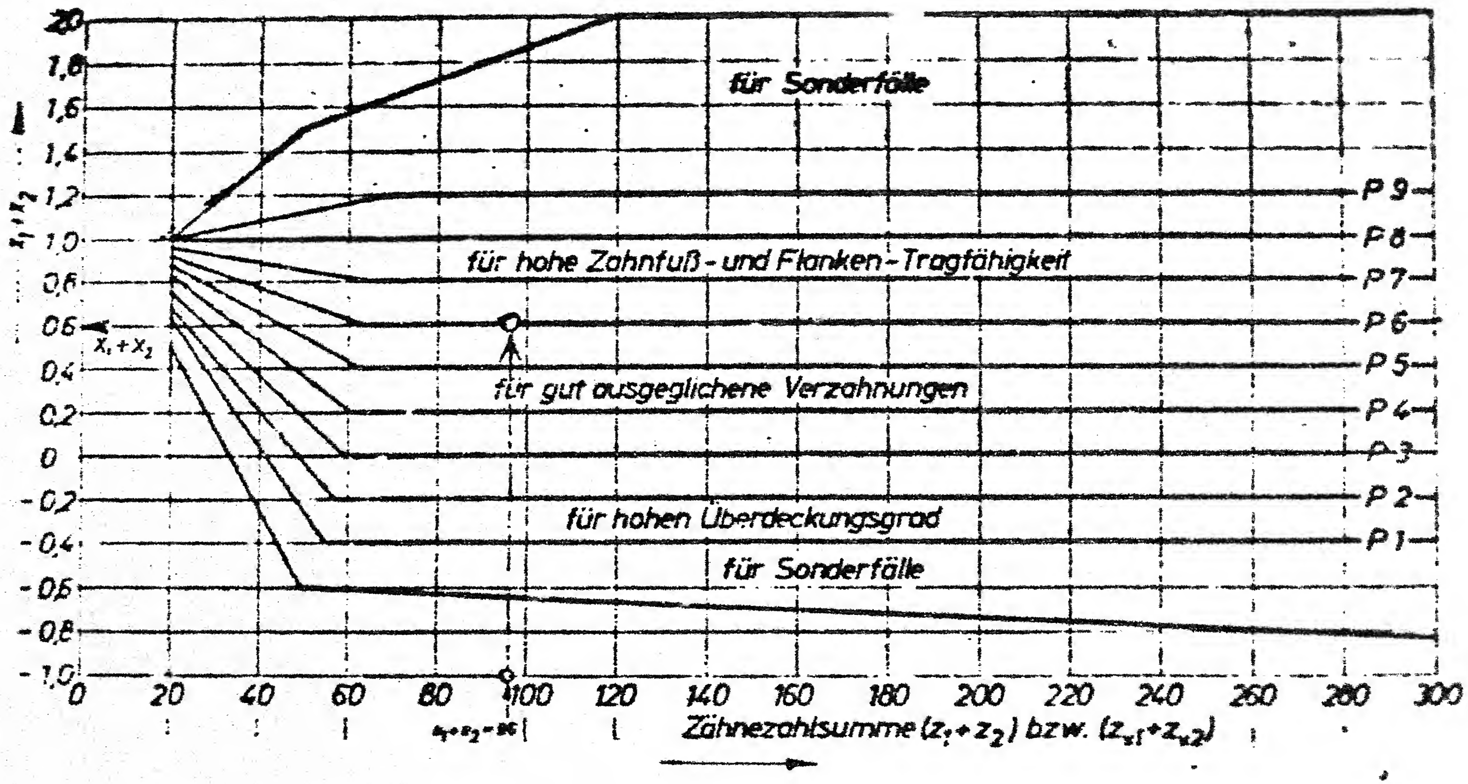


FIG. 4.3 OVERVIEW FOR THE SELECTION OF $(X_1 + X_2)$ FROM DIN 3992

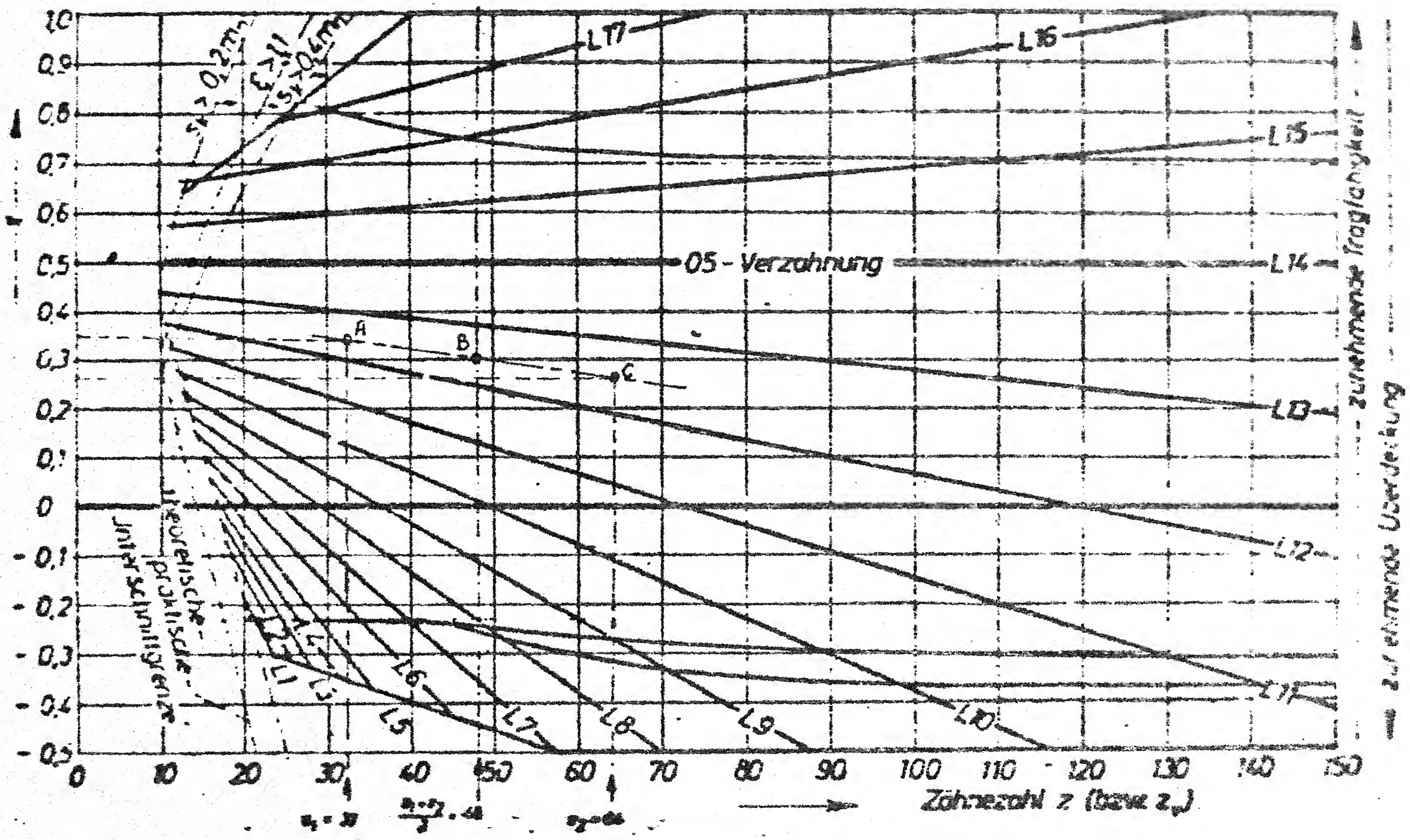


FIG. 4.4 OVERVIEW FOR THE SELECTION OF X_1 AND X_2 FROM DIN 3992

Step-1 : For the given operating conditions and the sum of the number of teeth on gear pair, find the total shift for all gear pairs of this stage using the chart given in Figure 4.3.

For an example to use the chart, assume

Z_1 = number of teeth on driver gear

= 32

Z_2 = number of teeth on driven gear

= 64

β = Helix angle on pitch circle

= 0°

For the balanced teeth condition use the line P6.

Here

$(Z_1 + Z_2) = 32 + 64 = 96$

Using plot P6 for balanced teeth condition, read the value of $(X_1 + X_2)$ opposite to $(Z_1 + Z_2)$.

Thus

$X = X_1 + X_2 = 0.6$

The difference to be noted here for the case of helical gears is, the value to be read on X-axis is the sum of equivalent number of teeth, $(Z_{v1} + Z_{v2})$, given by

Now α_b is given by

$$\text{inv } \alpha_b = \text{inv } \alpha_o + \frac{2\,(X_1 + X_2)\tan \alpha_{on}}{(Z_1 + Z_2)} \qquad \text{....} \quad (4.4)$$

with α_o from

$$\tan \alpha_o = \tan \alpha_{on}/\text{Cos}\beta \qquad (4.5)$$

and $\quad \text{inv } \alpha_o = \tan \alpha_o - \alpha_o \qquad (4.6)$

where

X_1 - profile correction factor on the driver gear

X_2 - profile correction factor on the driven gear.

Step-3: The corrected center distances calculated above are now averaged for each stage, and this average center distance is taken as the actual or final center distance 'a_m' for that particular stage. Now again based on this center distance the individual gear shifts are calculated with the help of chart shown in Figure 4.4 and the following formulation.

Based on corrected center distance, total shift is again calculated as

$$X_1 + X_2 = \frac{(\text{inv } \alpha_b - \text{inv } \alpha_o)(Z_1 + Z_2)}{2 \tan \alpha_{on}} \qquad \ldots\ldots \quad (4.7a)$$

with α_b from

$$\text{Cos } \alpha_b = \frac{a}{a_m} \text{ Cos } \alpha_o \qquad (4.7b)$$

Based on this $(X_1 + X_2)$ Calculated and $(Z_1 + Z_2)$, the individual corrections X_1 and X_2 are obtained from Figure 4.4. In this plot the values to be read on X-axis are $(Z_1 + Z_2)/2$ or $(Z_{v1} + Z_{v2})/2$ and on Y-axis are $(X_1 + X_2)/2$.

How to read X_1 and X_2 is explained with the following example:

Let

$$Z_1 = 32$$

$$Z_2 = 64$$

$$\frac{Z_1 + Z_2}{2} = 48$$

Let above calculated value of $(X_1 + X_2) = 0.6$

$$\left(\frac{X_1 + X_2}{2}\right) = 0.3$$

Read 48 on X-axis and 0.3 on Y-axis, thus we get point 'B' (See Figure 4.4). Now from point 'B' draw a line (shown dotted) in between two existing lines on the diagram, such that these

three intersect at a point. Now drop the perpendiculars from (Z_1 or Z_{v1}) and (Z_2 or Z_{v2}) to cut the line drawn at 'A' and 'C'. Read the values of X_1 and X_2 at Y-axis corresponding to the point 'A' and 'C' respectively.

Thus for the above example

$$X_1 = 0.34$$

$$X_2 = 0.26$$

In the present work separate subroutines are written to convert these two charts into logical program. The only input required from the designer is the choice of region for the chart shown in Figure 4.3.

4.3 Gear Inspection Data

Gear inspection data are the same important gear dimensions which are to be specified on the drawing for the production and inspection purposes.

In the present work equations used to calculate these data are taken from [3,17].

As inspection data are referred to a particular gear, in the present Chapter symbols are used without referring to any stage or gear pair. Throughout the discussion suffix 1 is used for the driver gear and 2 for the driven gear. Formulas given are valid for involute

gearing, assuming profile shifted gears.

Following are the inspection data calculated in the present program:

1) Pitch Circle Diameter (d)

$$d_1 = \frac{Z_1 . m}{Cos\beta}$$
$$d_2 = \frac{Z_2 . m}{Cos\beta} \tag{4.8}$$

2) Tip Circle Diameter (d_k)

$$d_{k_1} = 2\ (a_m + m - mX_2) - d_2$$
$$d_{k_2} = 2\ (a_m + m - mX_1) - d_1 \tag{4.9}$$

where

a_m - final center distance after gear corrections

m - normal module.

3) Base Circle Diameter (d_g)

$$d_{g_1} = d_2 \ Cos\ \alpha_{no}$$
$$d_{g_2} = d_2 \ Cos\ \alpha_{no} \tag{4.10}$$

4) Root Circle Diameter (d_f)

$$d_f = (d + 2mX - 2m) - 2S_k$$

where

$$S_k = \text{bottom clearance}$$
$$= 0.2m$$

$$\therefore \quad d_{f_1} = d_1 + 2mX_1 - 2.4m$$

$$d_{f_2} = d_2 + 2mX_2 - 2.4m \qquad (4.11)$$

5) Rolling Circle Diameter (d_b)

$$d_{b_1} = 2\, a_m \frac{Z_1}{Z_1 + Z_2}$$

$$d_{b_2} = d_{b_1} \frac{Z_2}{Z_1} \qquad (4.12)$$

6) Roller Diameter for the Measurement of Gears (d_r)
(Figure 4.5)

$$d_r = 2 \cdot r_r$$

$$d_r = \frac{\overline{I}_o}{\text{Cos } \alpha_{SM}}$$

$$d_r = \frac{d \text{ Sin } \lambda_o}{\text{Cos } (\alpha_o + \lambda_o)}$$

$$d_r = \frac{d \text{ Sin } \frac{\pi}{2Z}}{\text{Cos } (\alpha - \frac{\pi}{2Z})} \qquad (4.13)$$

The roller diameter obtained by the above formula is to be rounded off to nearest tenth of millimetre. This

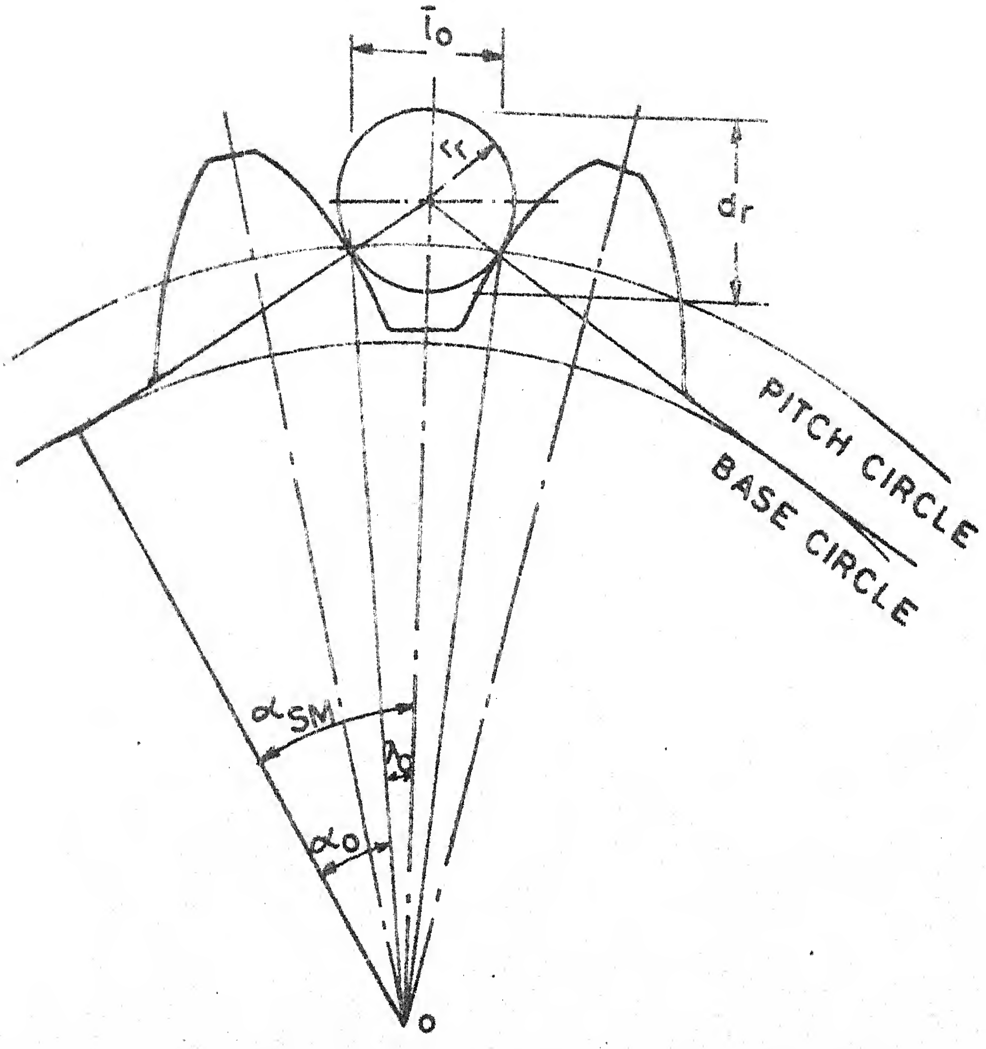


FIG. 4·4 ROLLER MEASUREMENT FOR EXTERNAL GEAR

rounded off value of roller diameter is to be used in further calculations.

7) <u>Over Roller Reading (M_i)</u> (Figure 4.6)

For even No. of teeth

$$M_i = \frac{d \cdot \cos\alpha_o}{\cos \alpha_{SM}} + d_r \tag{4.14a}$$

For odd No. of teeth

$$M_i = \frac{d \cdot \cos\alpha_o}{\cos \alpha_{SM}} \cdot \cos \frac{90°}{Z} + d_r \tag{4.14b}$$

where α_{SM} is given by

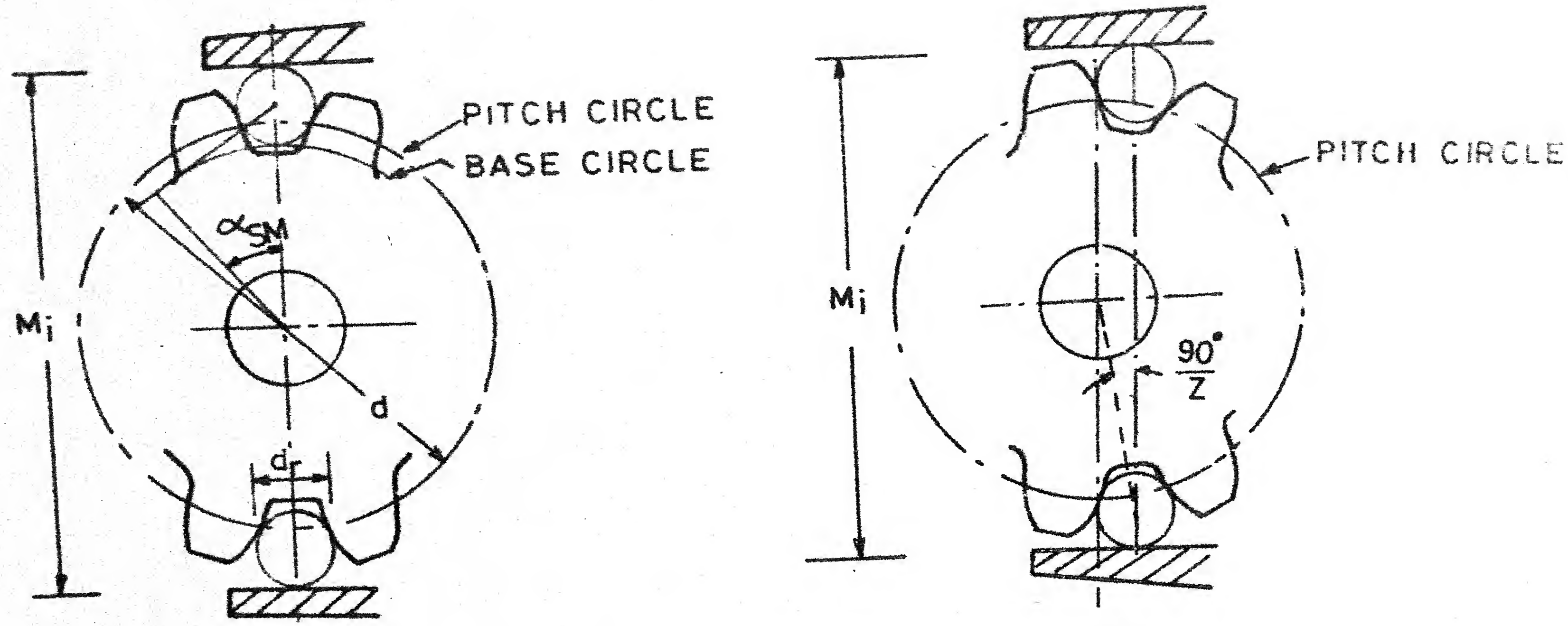
$$\text{inv. } \alpha_{SM} = \text{inv. } \alpha_o + \frac{d_r}{m. Z. \cos\alpha_o} + \frac{2. X. \tan \alpha_o}{Z} - \frac{\pi}{2.Z} + \frac{A_w}{m.Z. \cos\alpha_o} \tag{4.14c}$$

where

X - corresponding gear profile shift factor

A_w - width over teeth allowance.

There are two values of A_w, namely, A_{ow} (upper allowance) and A_{uw} (lower allowance). Thus find out two values of α_{SM} which will respectively give to two limits for over roller reading. Here A_{ow} and A_{uw} are the inputs from user.



(a) EVEN NO. OF TEETH

(b) ODD NO. OF TEETH

FIG. 4.5 MEASUREMENT OF OVER ROLLER READING

CHAPTER-5

PROGRAMMING CONSIDERATIONS AND EXAMPLES

The present program has been written in Fortan-10, developed and tested on DEC-1090 system. In the present chapter the major programming considerations are put in the form of flow charts Figures 5.1 - 5.5, also the purpose of different subroutines is explained.

5.1 The Main Program

The purpose of main program is to accept the input data interactively to calculate some of the parameters in itself and others by calling subroutines and lastly to print the results. Two separate files are opened within the program, one to store the data to be used in Graphic subroutine and the other to store all the questions asked as well as their answers. Second file can be printed to get a permanent record.

The program is written in a very general way, so that it can handle number of different cases. Most of the interaction dialogues are built in the main program.

Separate subroutines are written for the following purposes:

1. Subroutine Corsum:

Purpose - To find the appropriate total correction factor 'X' according to the chart shown in Figure 4.3.

2. Subroutine Cendis:

Purpose - To calculate the corrected center distance based on the above total correction factor.

3. Function Calalf:

Purpose - To solve a transcendental equation

4. Subroutine Shift:

Purpose - To calculate total profile shift factor based on corrected centerdistances.

5. Subroutine Corind:

Purpose - To find individual corrections according to chart given in Figure 4.4

6. Subroutine Roller:

Purpose - To calculate over roller reading

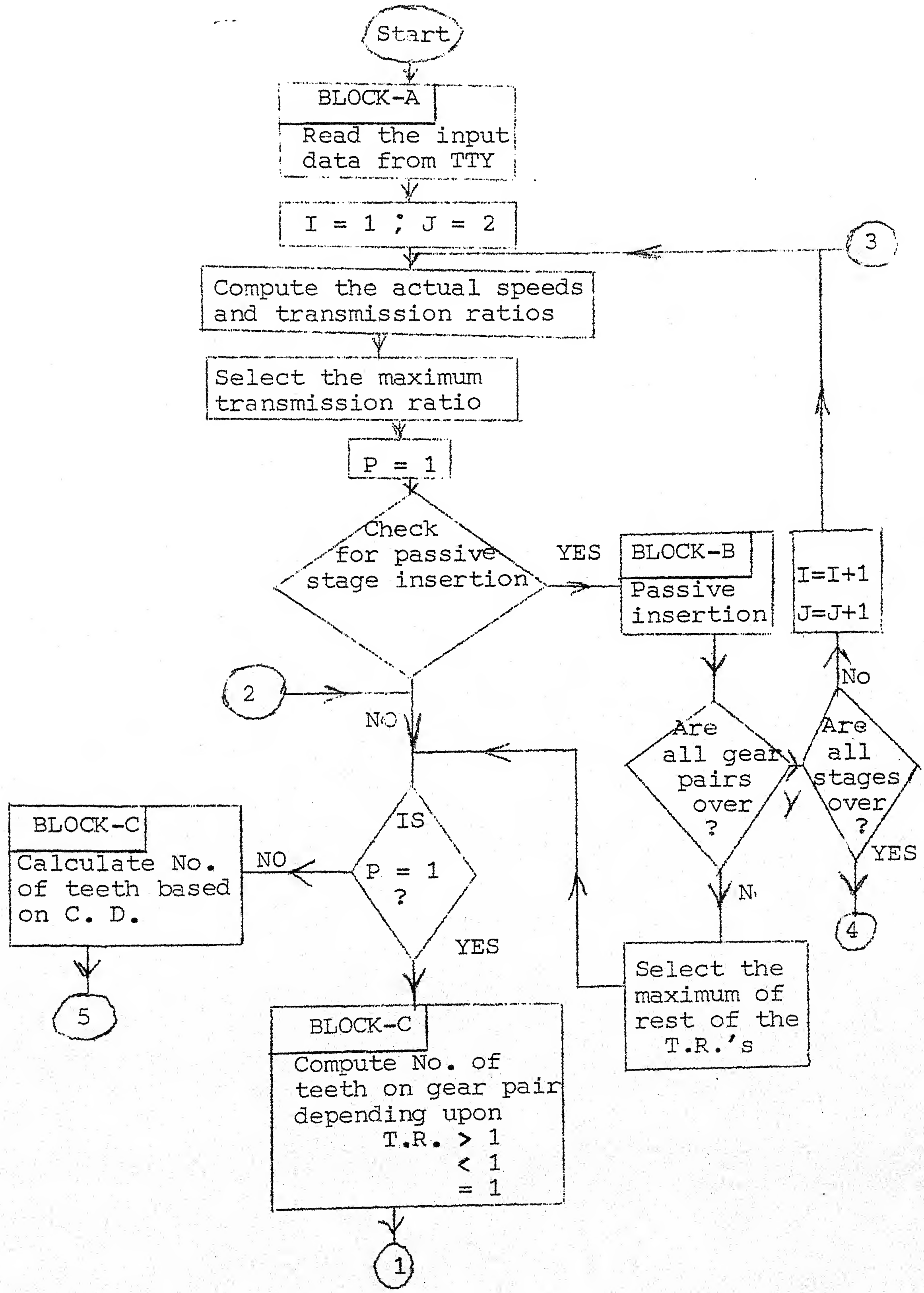
7. Subroutine Teeth1:

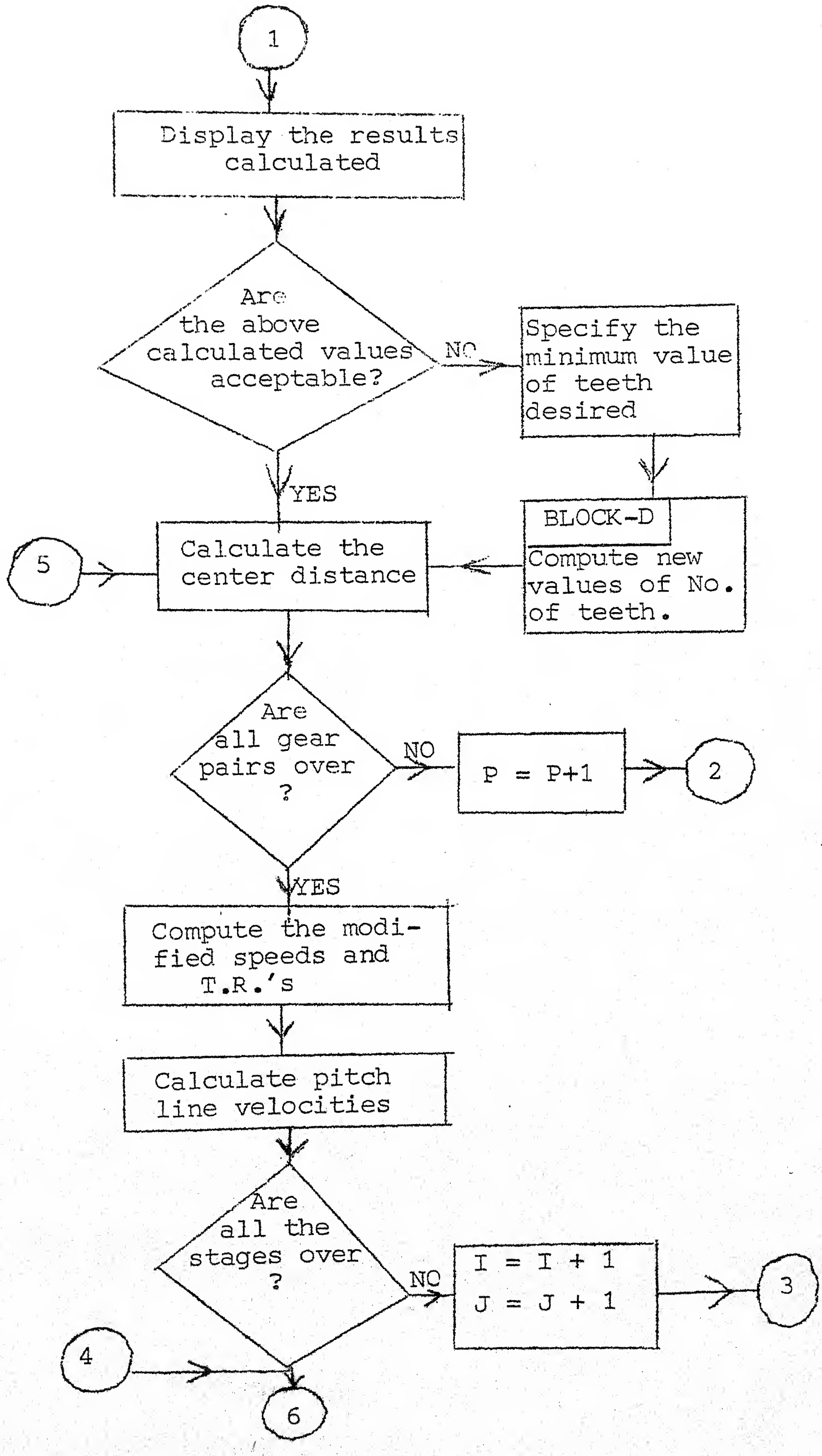
Purpose - To calculate the number of teeth when transmission ratio is less than 1. Also make interaction with the user.

8. Subroutine Teeth2:

Purpose - To calculate number of teeth when transmission ratio is more than 1.

<u>Master Flow Chart</u>




1
Display the results calculated
Are the above calculated values acceptable?
NO
Specify the minimum value of teeth desired
YES
BLOCK-D
Compute new values of No. of teeth.
5
Calculate the center distance
Are all gear pairs over ?
NO
P = P+1
2
YES
Compute the modified speeds and T.R.'s
Calculate pitch line velocities
Are all the stages over ?
NO
I = I + 1
J = J + 1
3
4
6

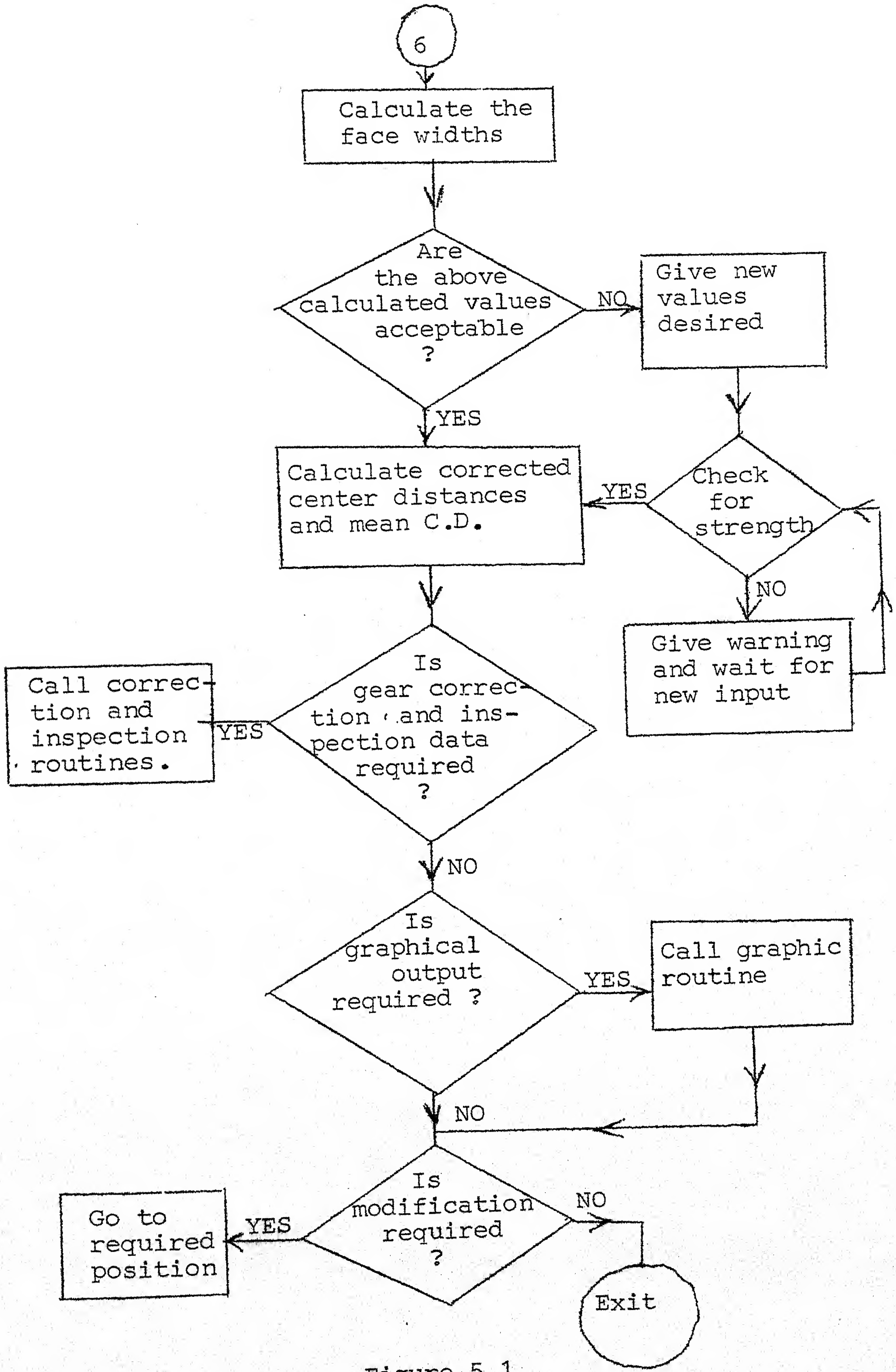

6
Calculate the face widths
Are the above calculated values acceptable ?
NO
Give new values desired
YES
Calculate corrected center distances and mean C.D.
YES
Check for strength
NO
Give warning and wait for new input
Call correction and inspection routines.
YES
Is gear correction and inspection data required ?
NO
Is graphical output required ?
YES
Call graphic routine
NO
Go to required position
YES
Is modification required ?
NO
Exit

Figure 5.1

Interactive Loop for Input Data

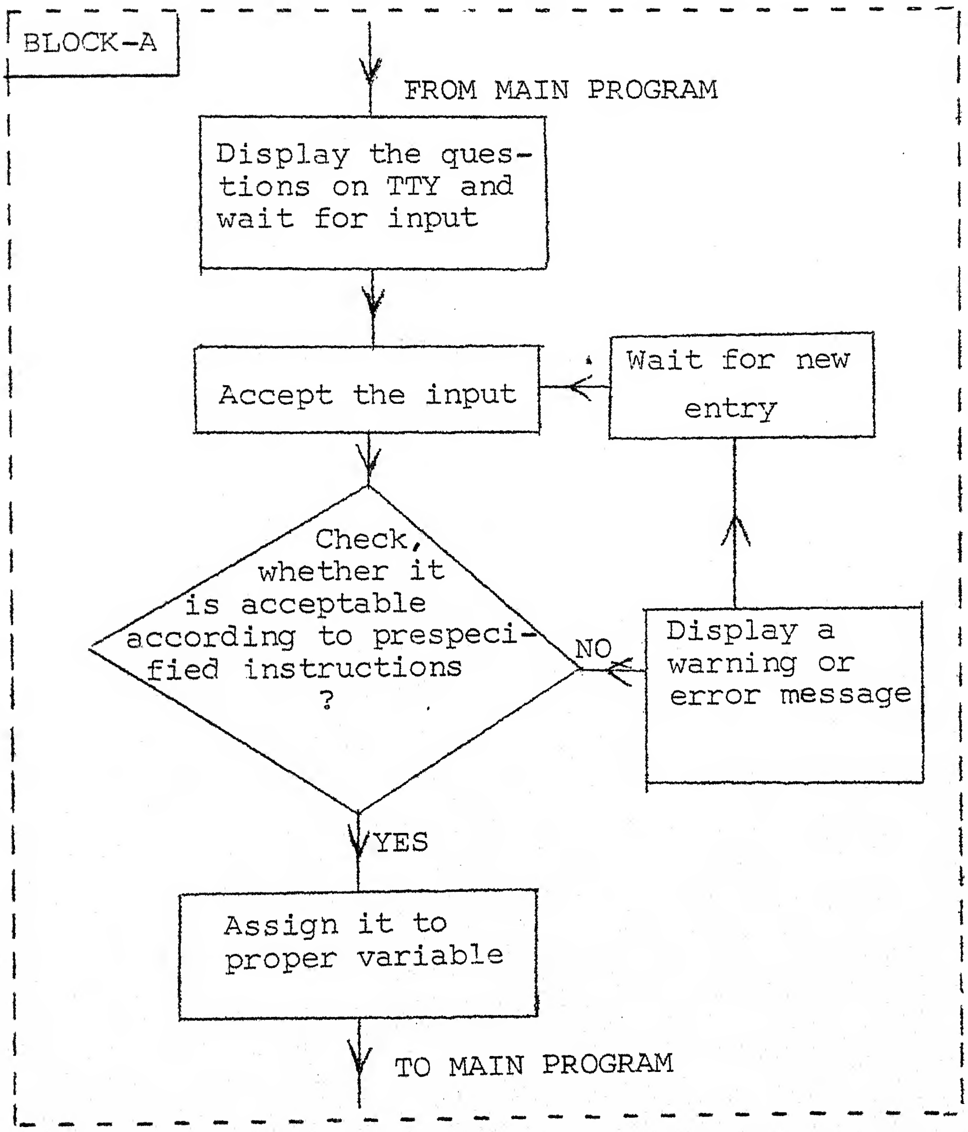


Figure 5.2

Flow Chart for Passive Insertion

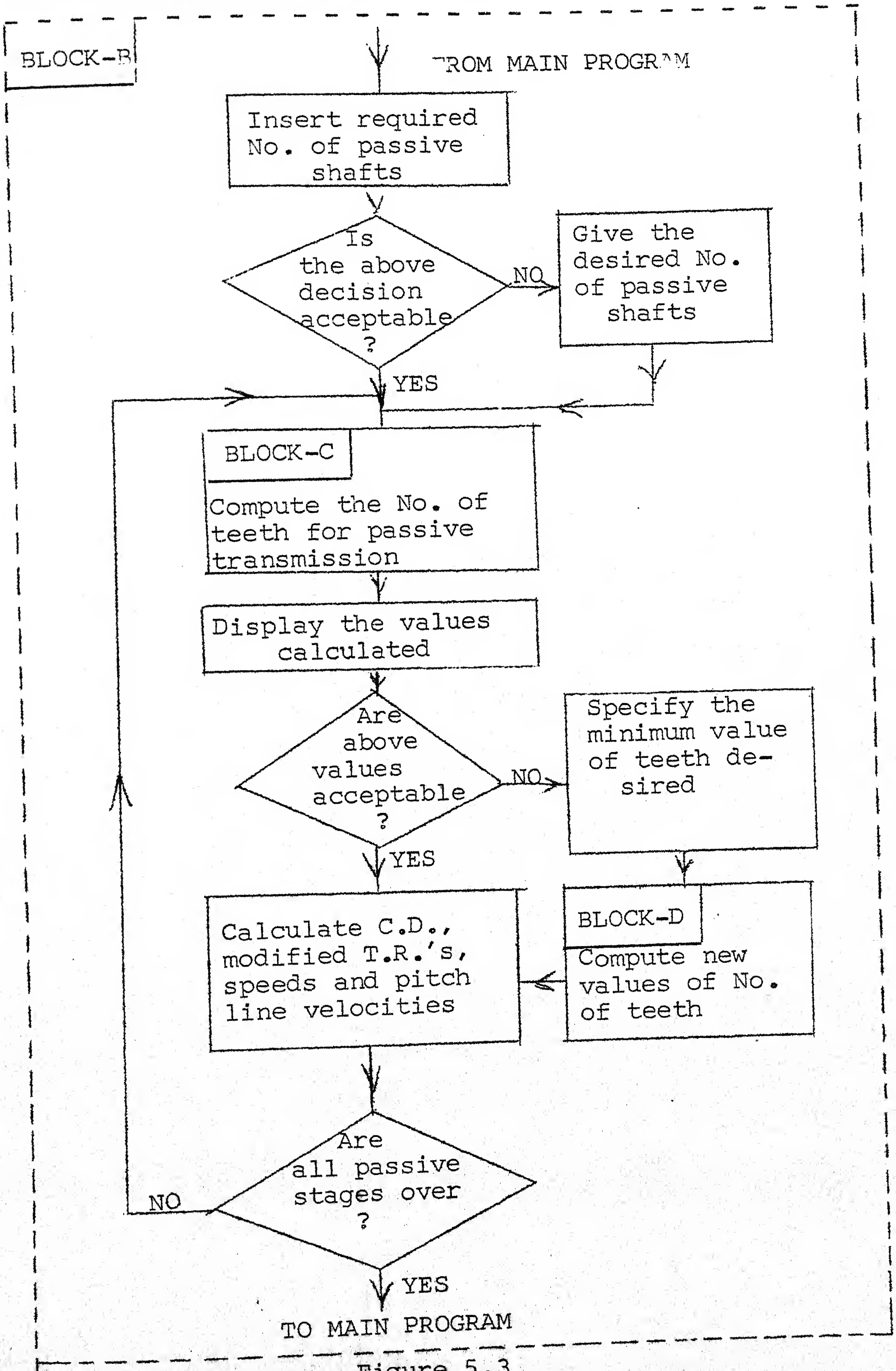


Figure 5.3

Flow Chart for Number of teeth Selection

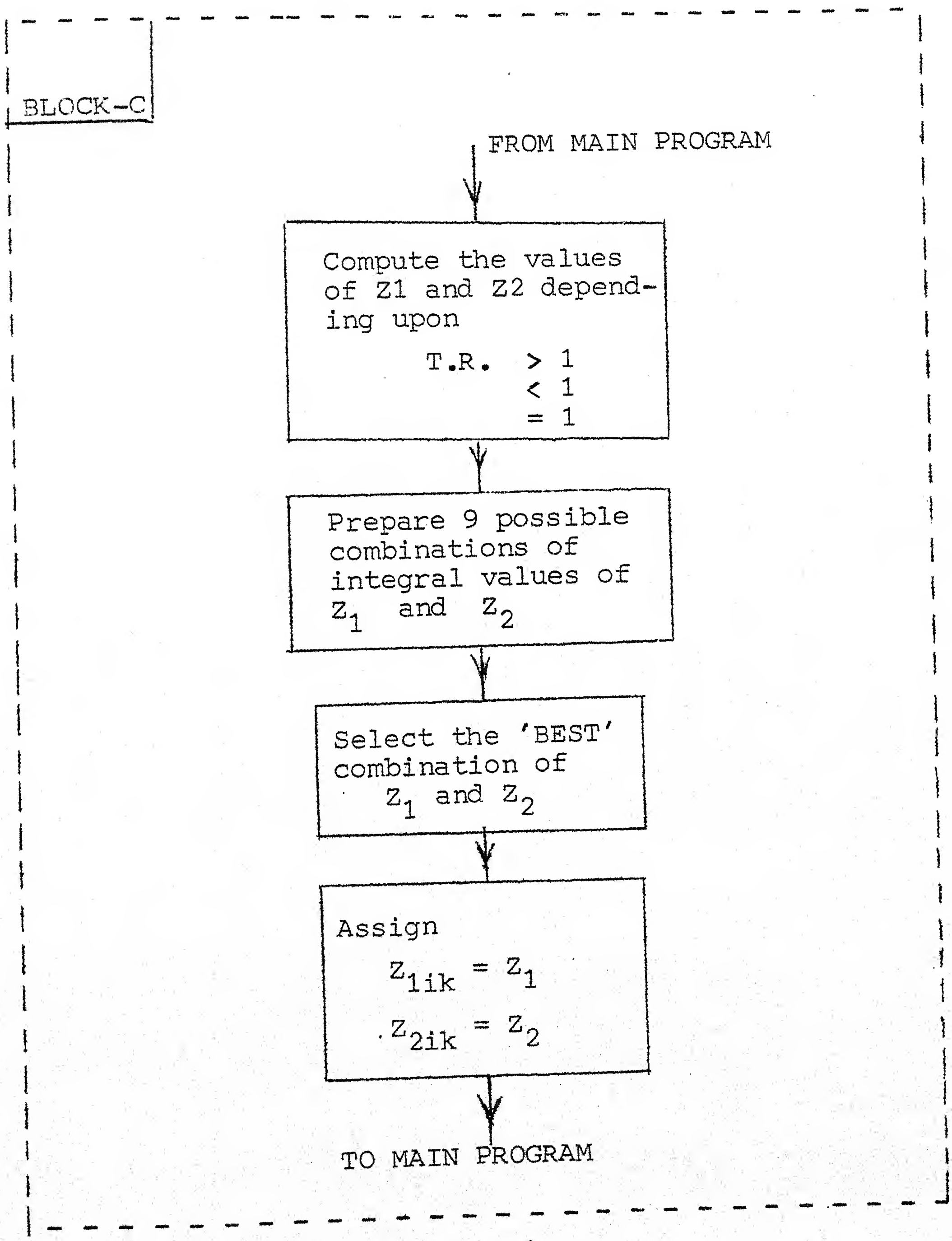


Figure 5.4

Flow Chart for Teeth Selection, when one of the Values is given by Designer

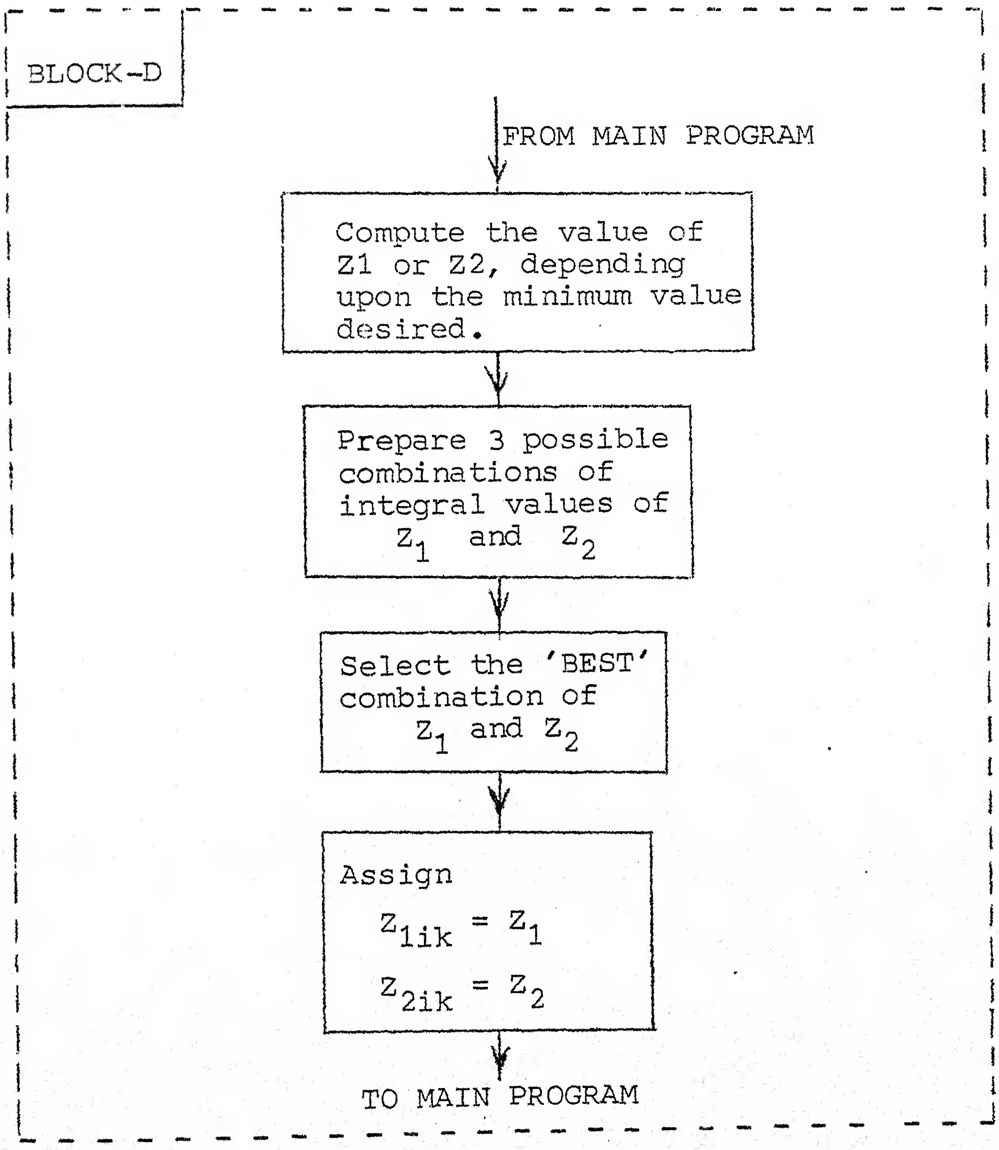


Figure 5.5

9. Subroutine Teeth3:

Purpose - To select the 'Best' combination of integer number of teeth.

10. Subroutine Maxi:

Purpose - To find minimum of real numbers.

11. Subroutine Graph:

Purpose - This subroutine opens the data file prepared in main program and according to that draw the graphic outputs.

5.2 Example No. 1 for 6-Speed Gearbox

In order to illustrate the design approach discussed in Ghapters 2, 3 and 4 with the help of the program developed, an example of a 6-speed gearbox is chosen. The input data as well as the intermediate changes made for the first case are given in Appendix-I.1. In this case all the possible structure diagrams as shown in Figures 2.2 have been taken into account and finally two structures have been selected. These are 1 X 3 X 2 and 1 X 2 X 3 respectively. In each case one can have either an open or closed structure. The output results in the form of speed diagrams ,the line diagrams and the pitch line velocity diagrams are shown in Figures 5.6 - 5.11.

5.3 Example No. 2 for 18-Speed Gearbox

This example has been selected to illustrate how the discontinuities in the speeds of any intermediate shaft can be taken into account. The input specifies the number of intermediate shaft to be 3. It should be noted that discontinuities in speed can occur only on the second and for the third intermediate shaft. In the present case 5 alternate structures have been explore as shown in Figures 5.12, 5.15 - 5.18. In each case complete design can be carried out and results can be printed in tabular form. For one representative case shown in Fig. 5.12, the outputs in the form of the line diagram and the pitch line velocity diagram are given in Figures 5.13 and 5.14. Input data corresponding to first case is given in Appendix-I.2.

5.4 Example No. 3 for 24-Speed Gearbox

This example pertains to a 24-speed gearbox of a milling machine developed by a Machine Tool Manufacturer. In this case it has been found necessary to introduce the passive shafts in one of the stages,since the gear ratio was too low. This has been successfully accomplished through the program. The input data is included in Appendix-I.3. It has been found advantageous

in practice to increase the speeds of intermediate shaft-gears so that the gearbox can occupy less space and then bring the speeds down on the output spindle speeds.
In such cases sometimes it becomes necessary to introduce the passive shafts also when the number of intermediate shafts are large, then it is necessary to check, that the percent variation in the speeds of the output spindle shaft are within permissible limits. For this purpose a slight modification in the method of design is recommended and discussed, in section 3.5. The output in the form of ray diagram the line diagram and pitch line velocity diagram are given in Figures 5.19 - 5.21. A complete set of tabulated output showing the values of various design parameters is also given in Tables 5.1 - 5.5.

6 SPEED GEAR BOX

1X3X2

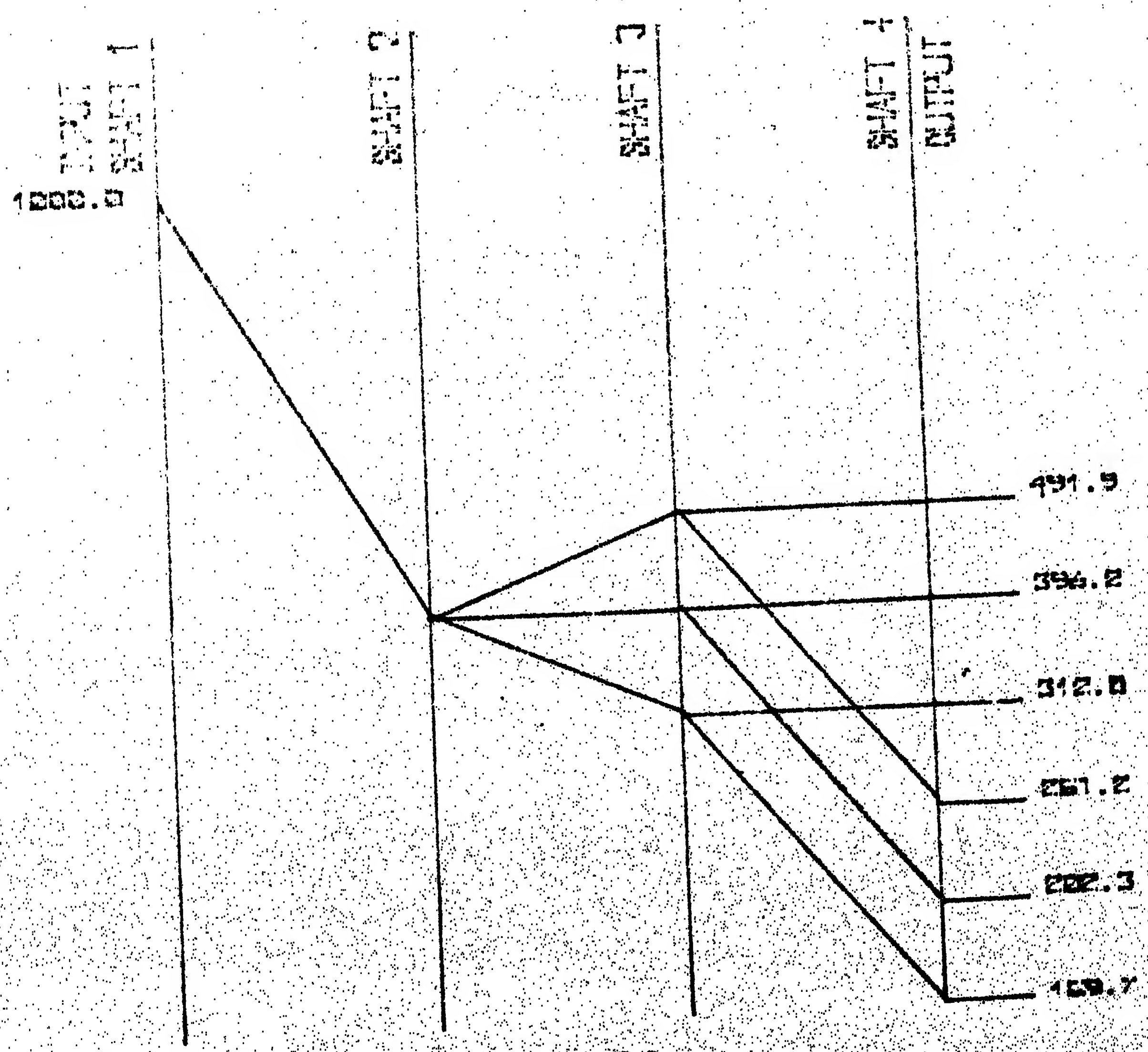


FIG.5.6

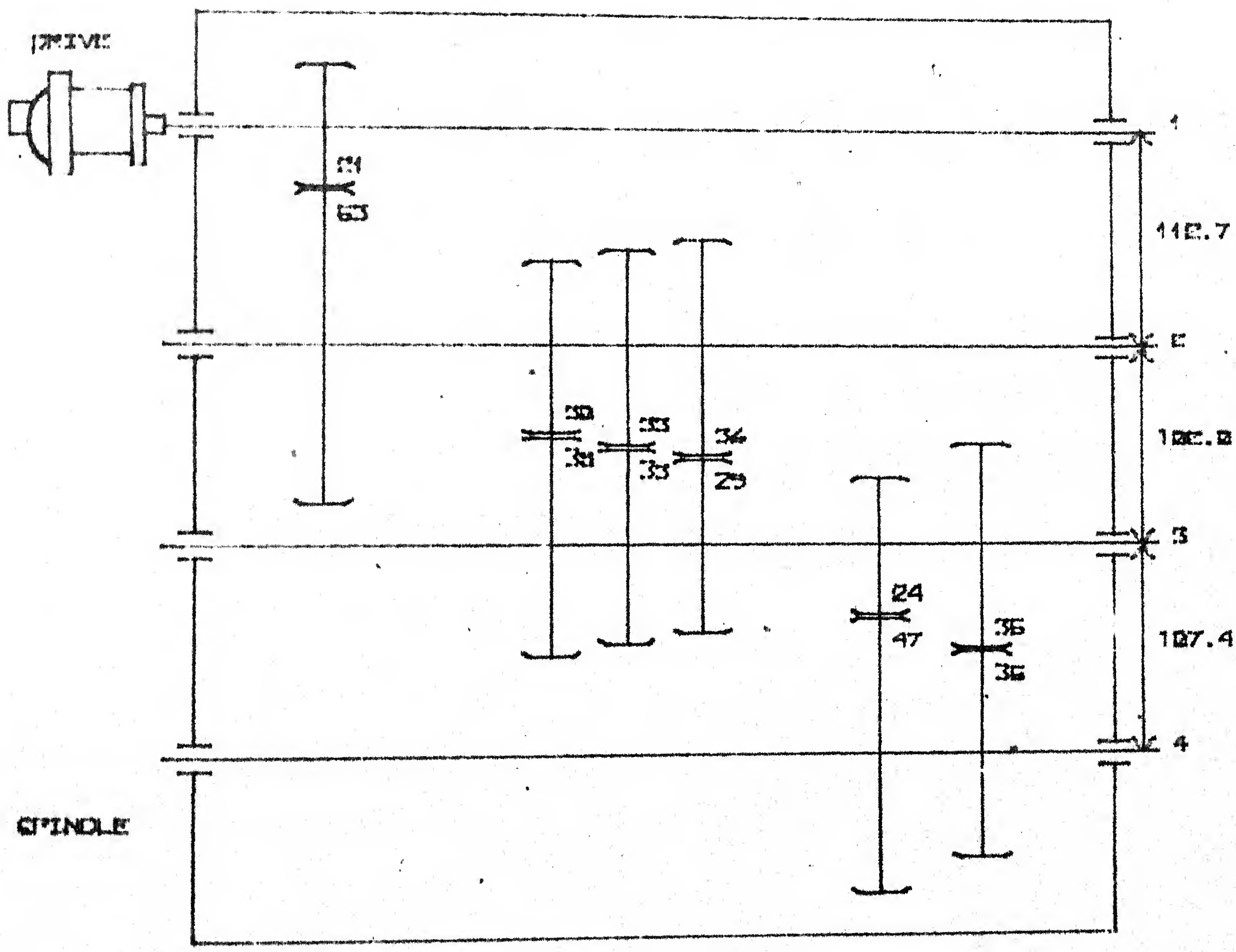


FIG.5.7 LINE DIAGRAM

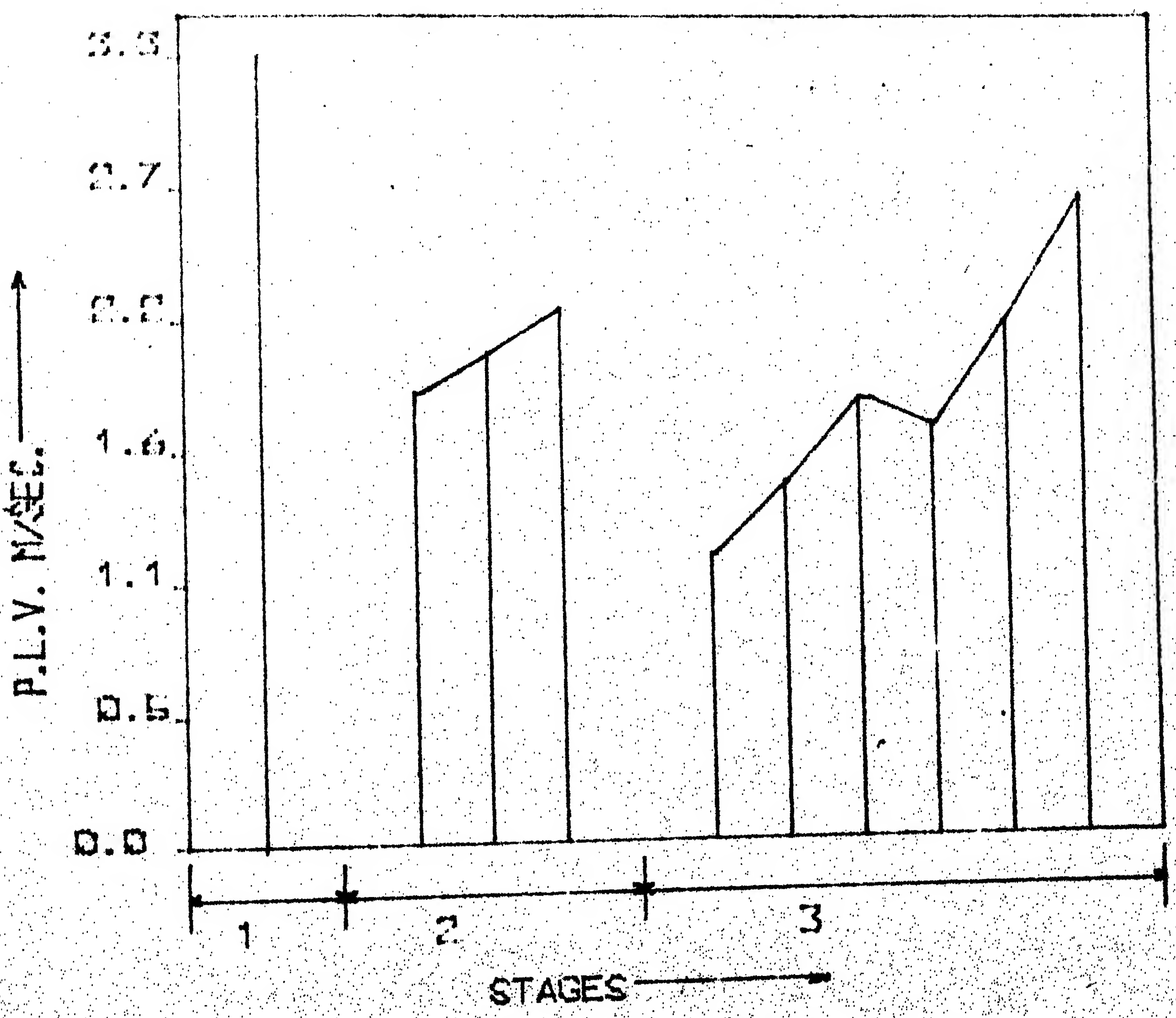


FIG.5.8 PITCH LINE VELOCITY DISTRIBUTION

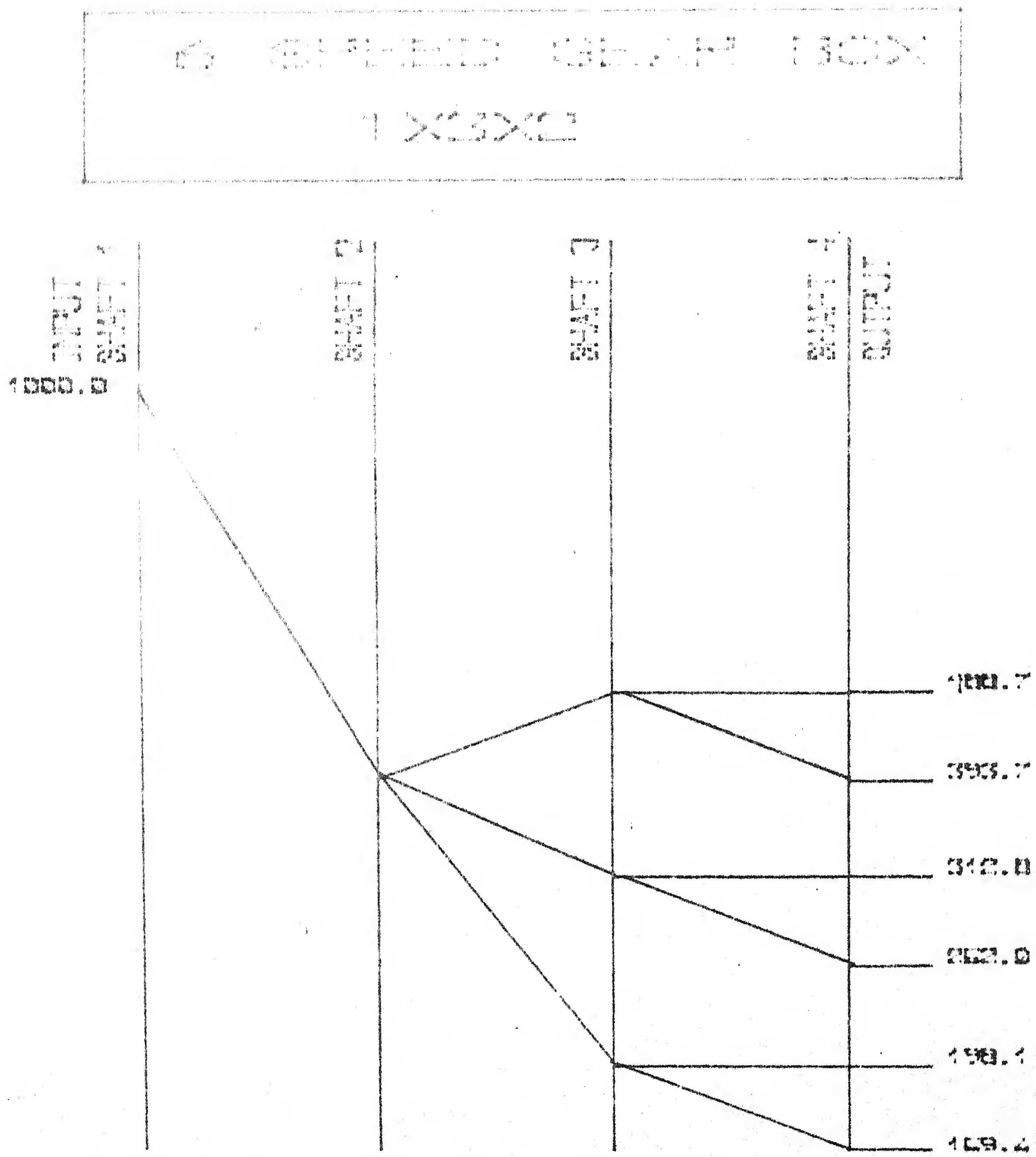

6 SPEED GEAR BOX
1X3X2
INPUT
SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
OUTPUT
1000.0

FIG.5·9

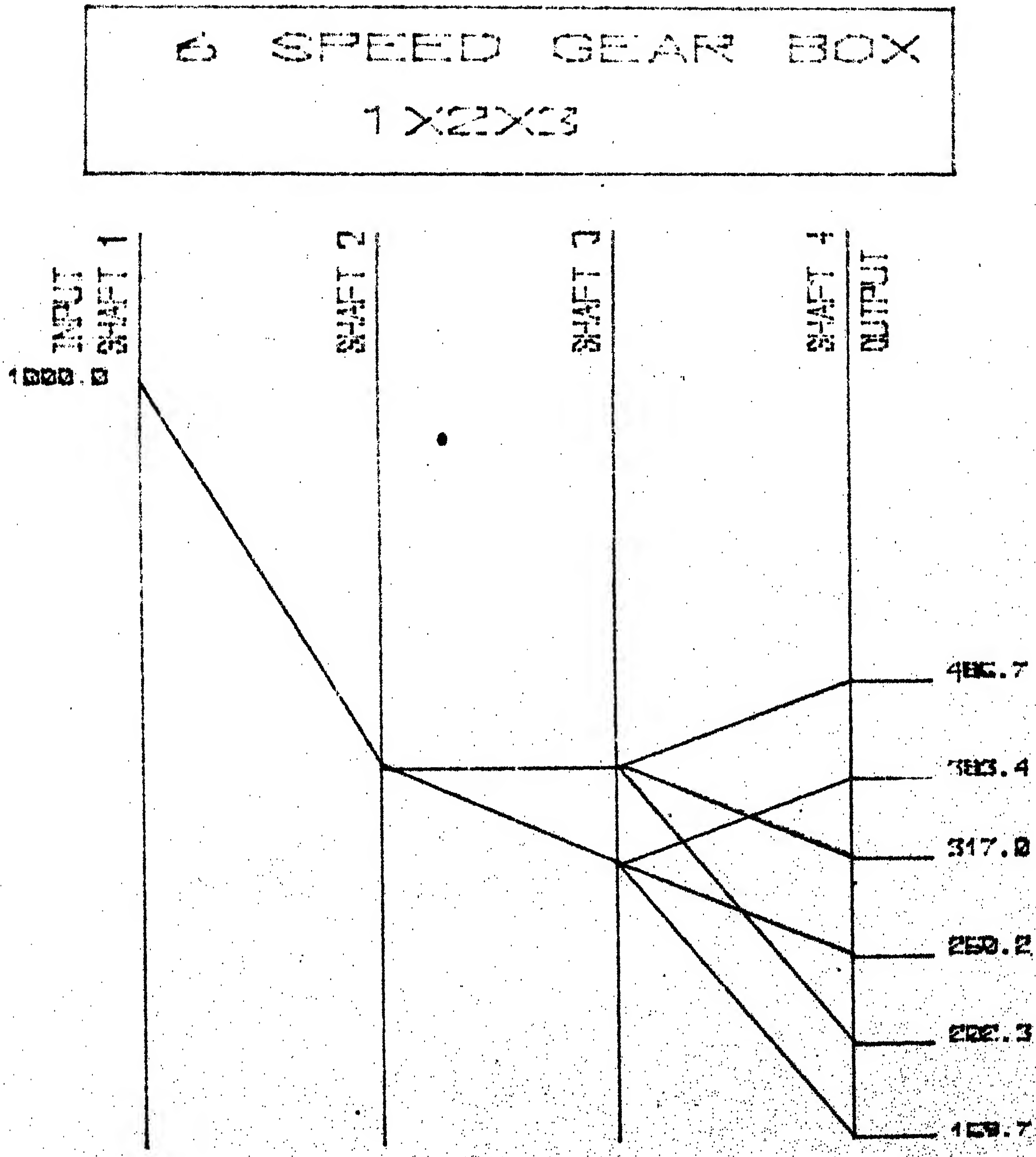

6 SPEED GEAR BOX
1X2X3
INPUT
SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
OUTPUT
1000.0
317.0
250.2
202.3

FIG. 5.10

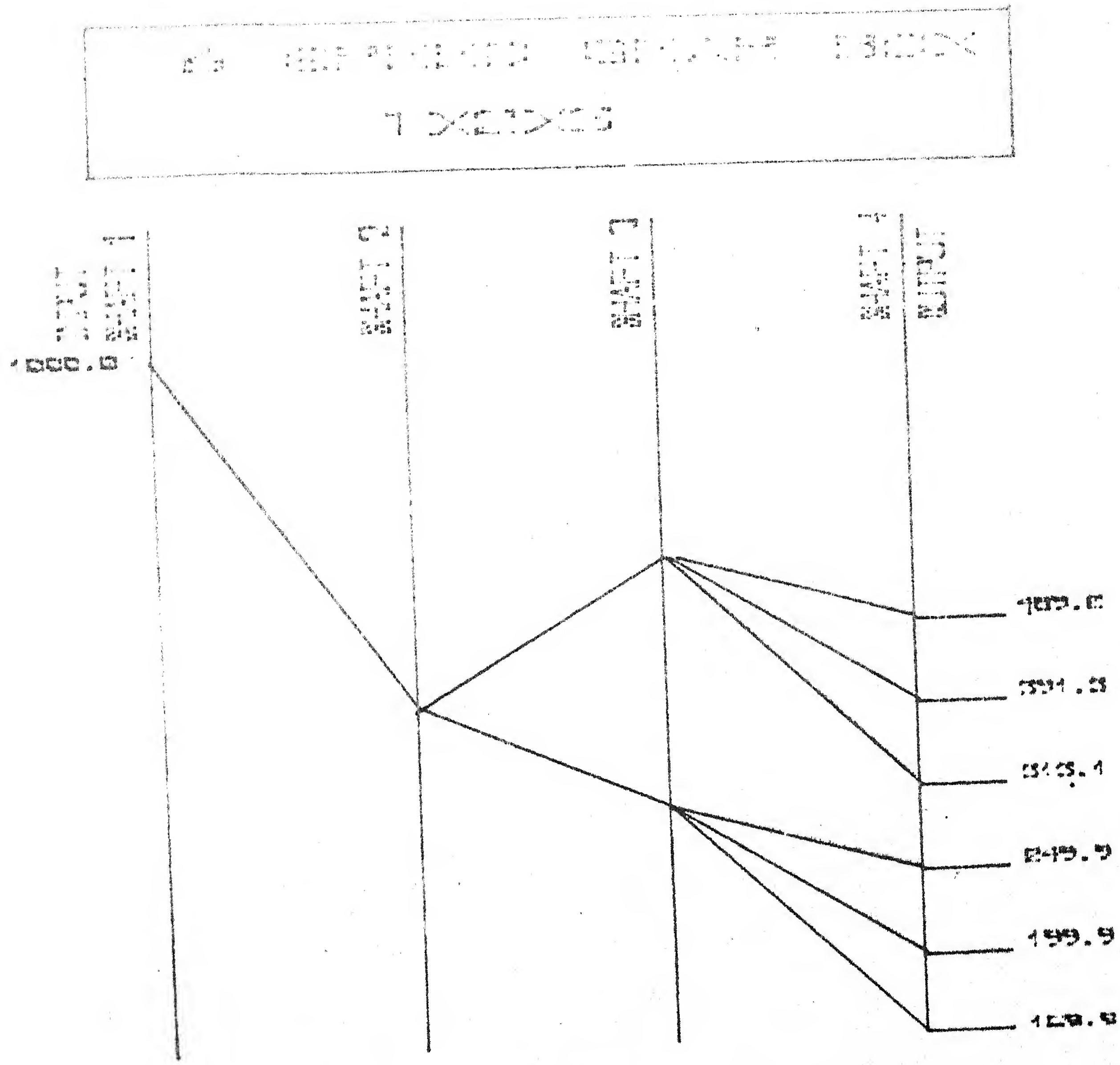


FIG. 5·11

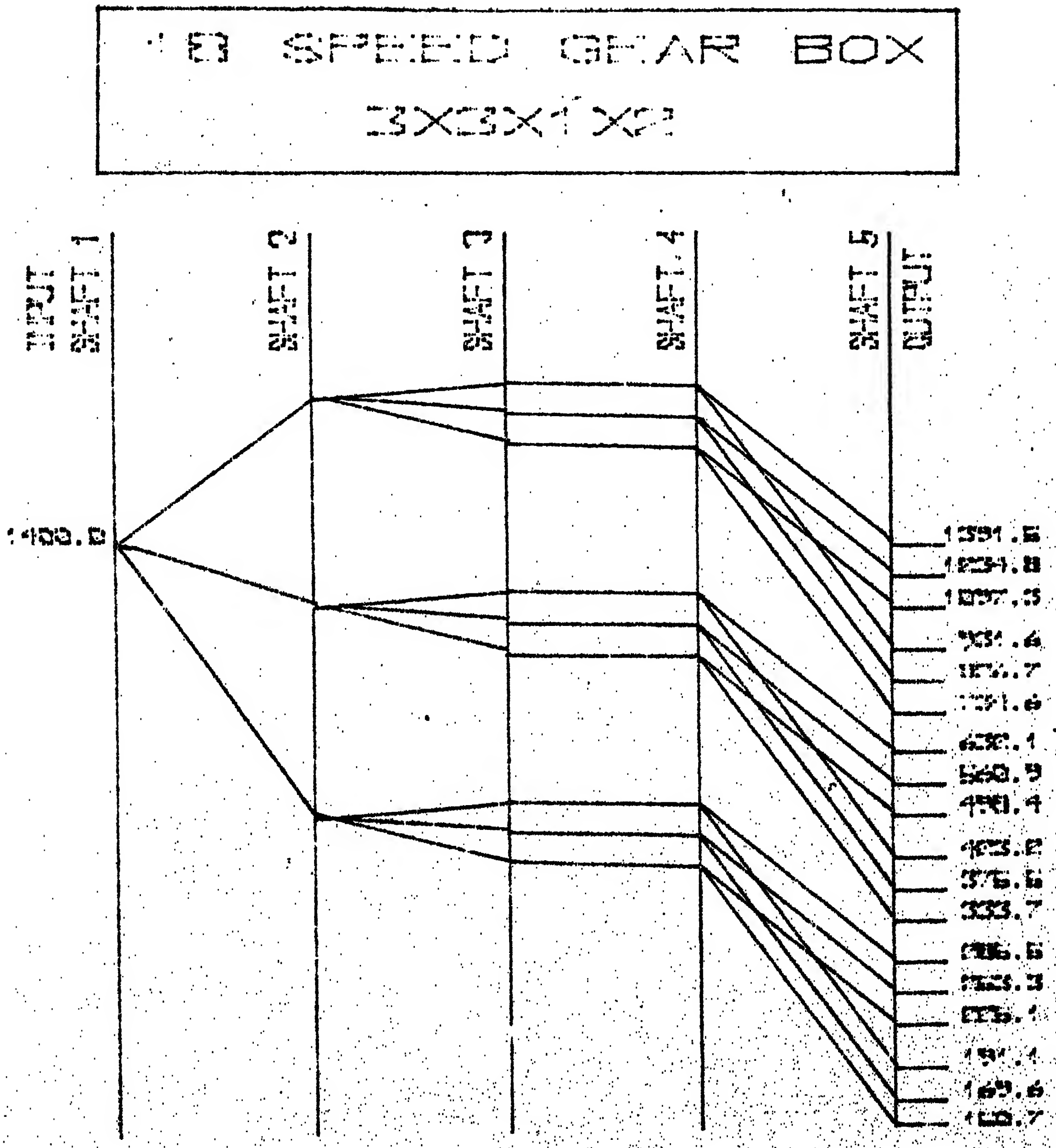

18 SPEED GEAR BOX
3X3X1X2
INPUT
SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
SHAFT 5
OUTPUT
1400.0
1391.5
560.9
333.7

FIG. 5·12

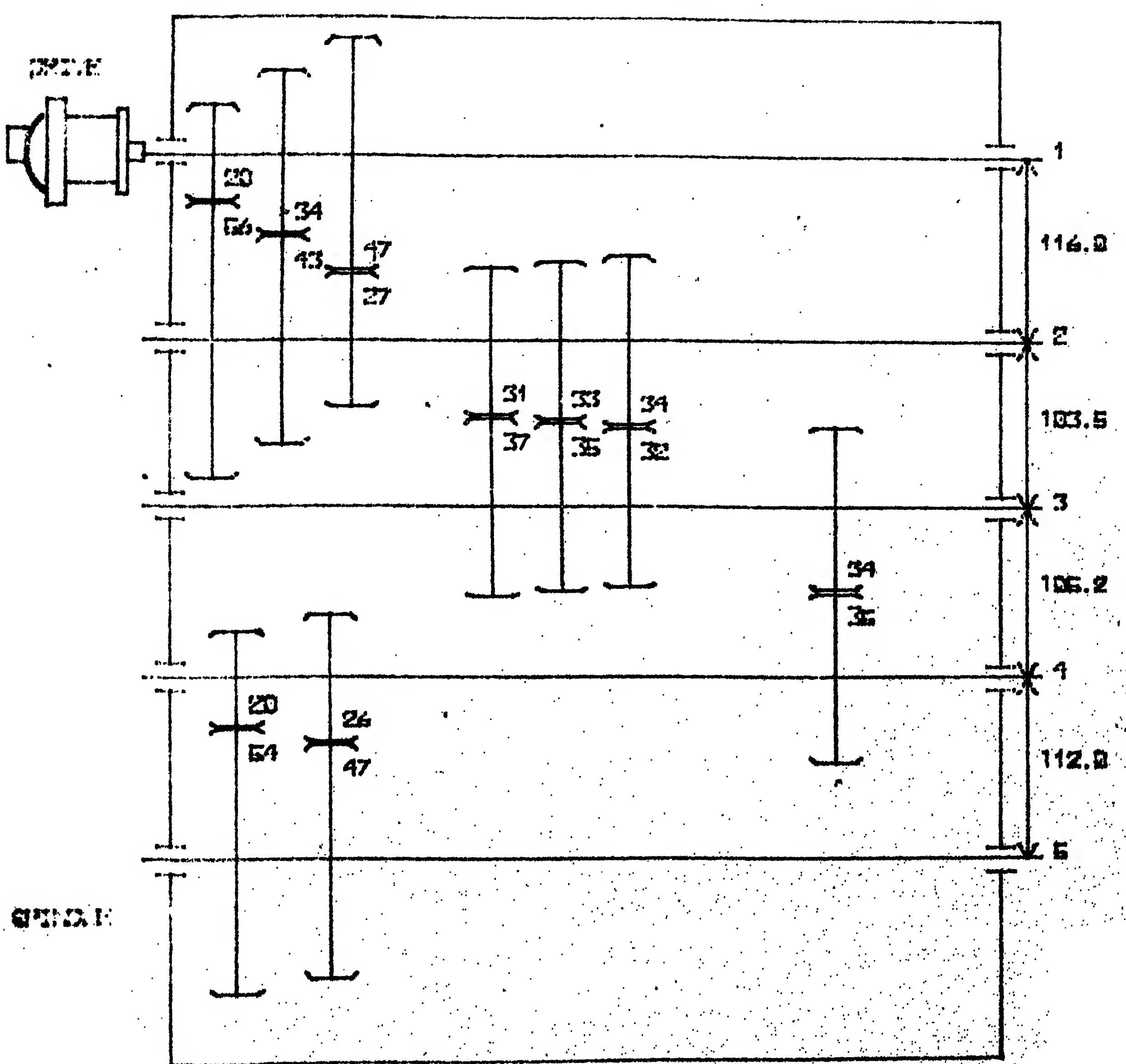

1
2
3
4
5
116.0
103.5
106.2
112.0
20
56
34
43
47
27
31
37
33
35
34
32
34
35
20
64
26
47

FIG.5.13

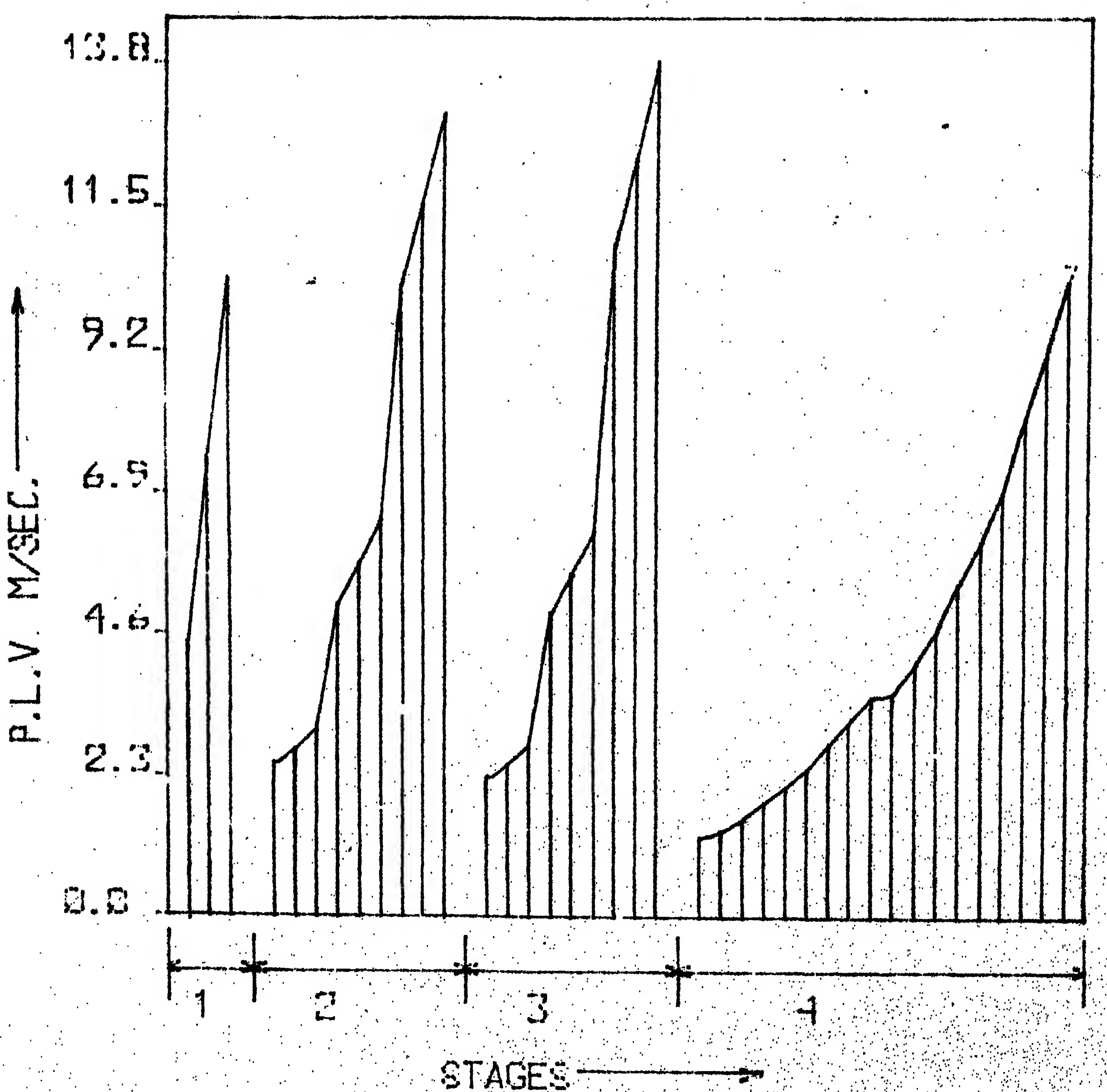


FIG. 5.14

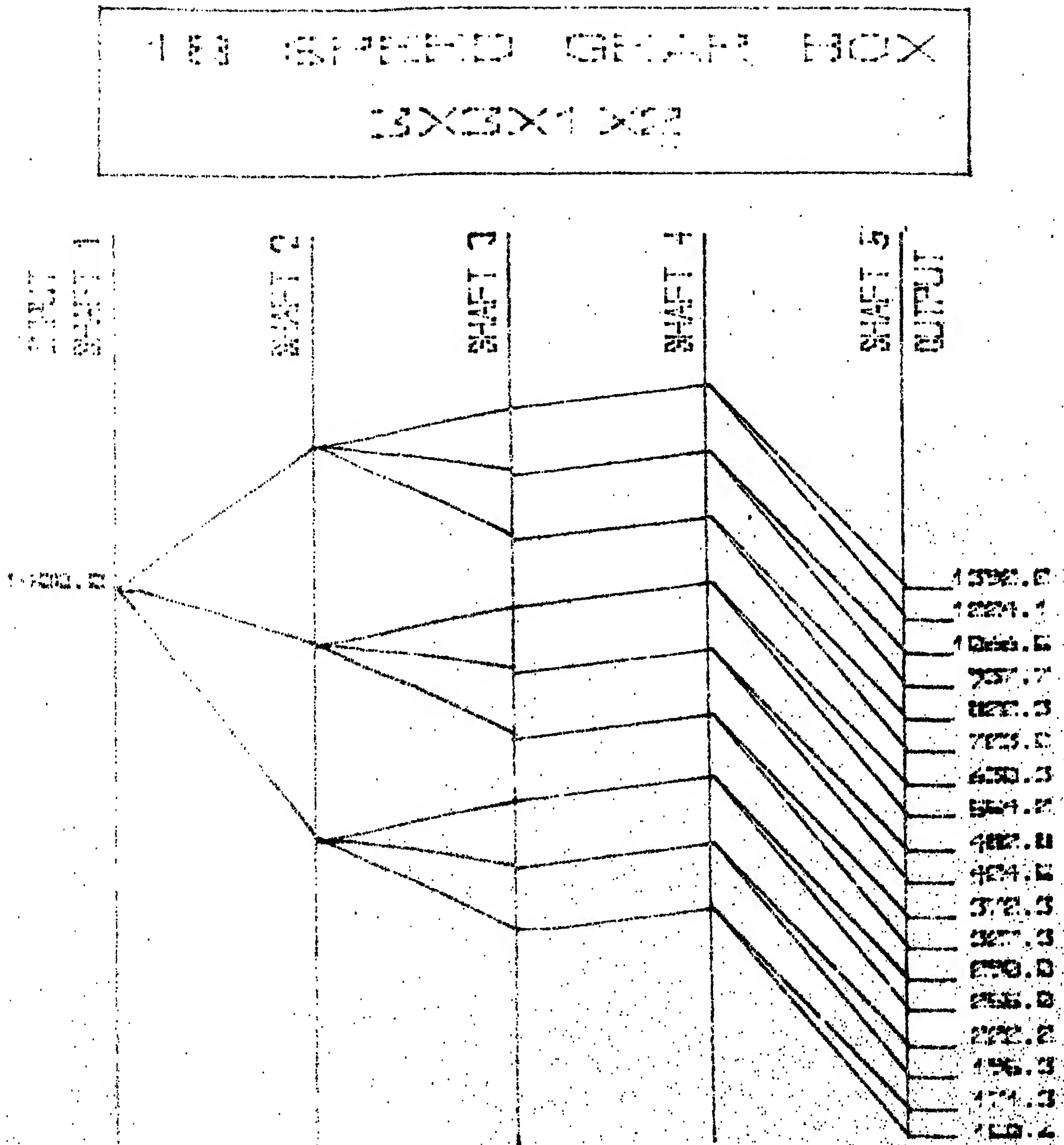


FIG 5.15

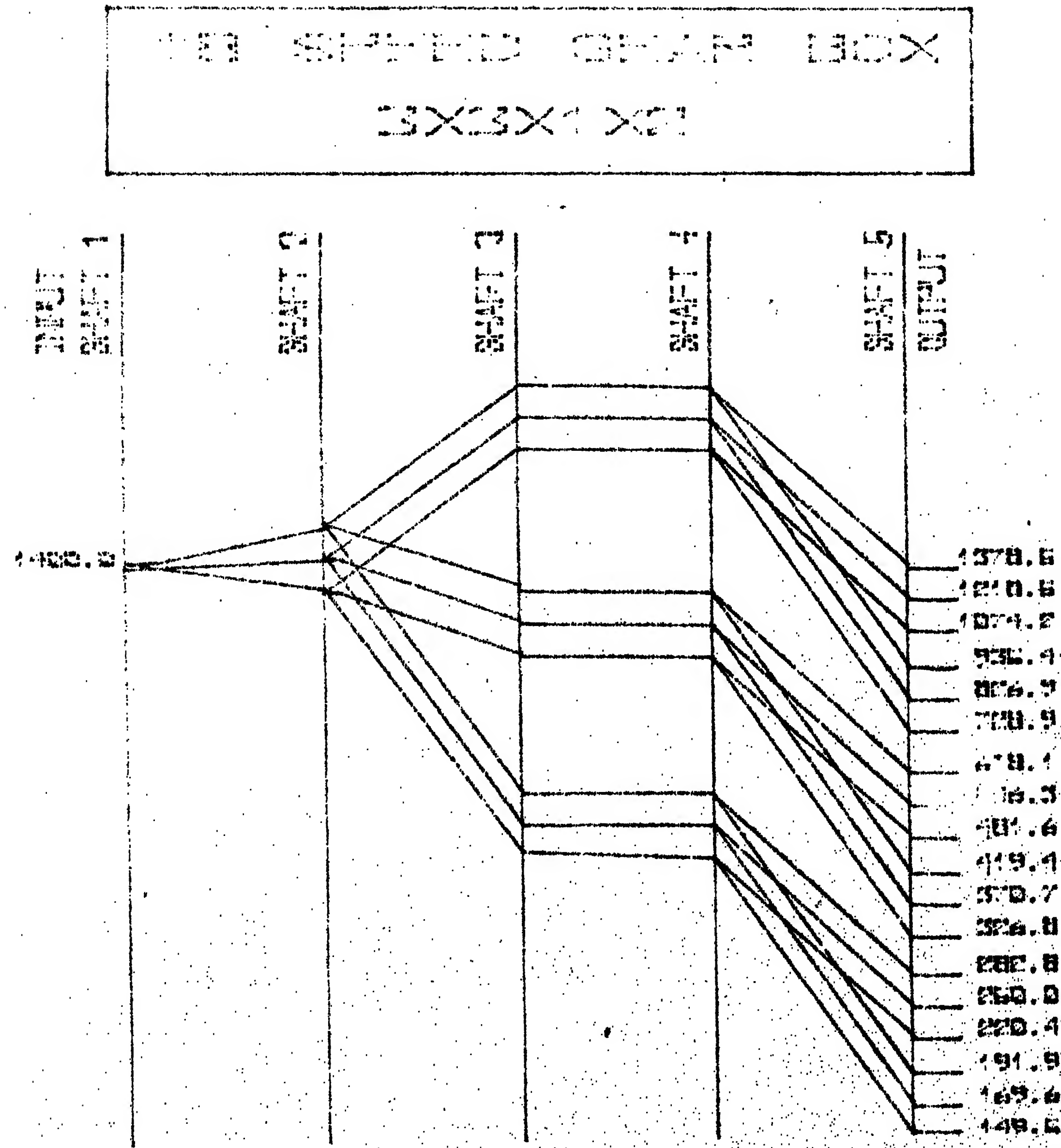

18 SPEED GEAR BOX
3X3X1X2
INPUT SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
SHAFT 5
OUTPUT
1400.0
1378.6
1074.2
419.4
370.7
326.8
282.8
250.0
220.4
191.9
169.6
149.6

FIG-5-16

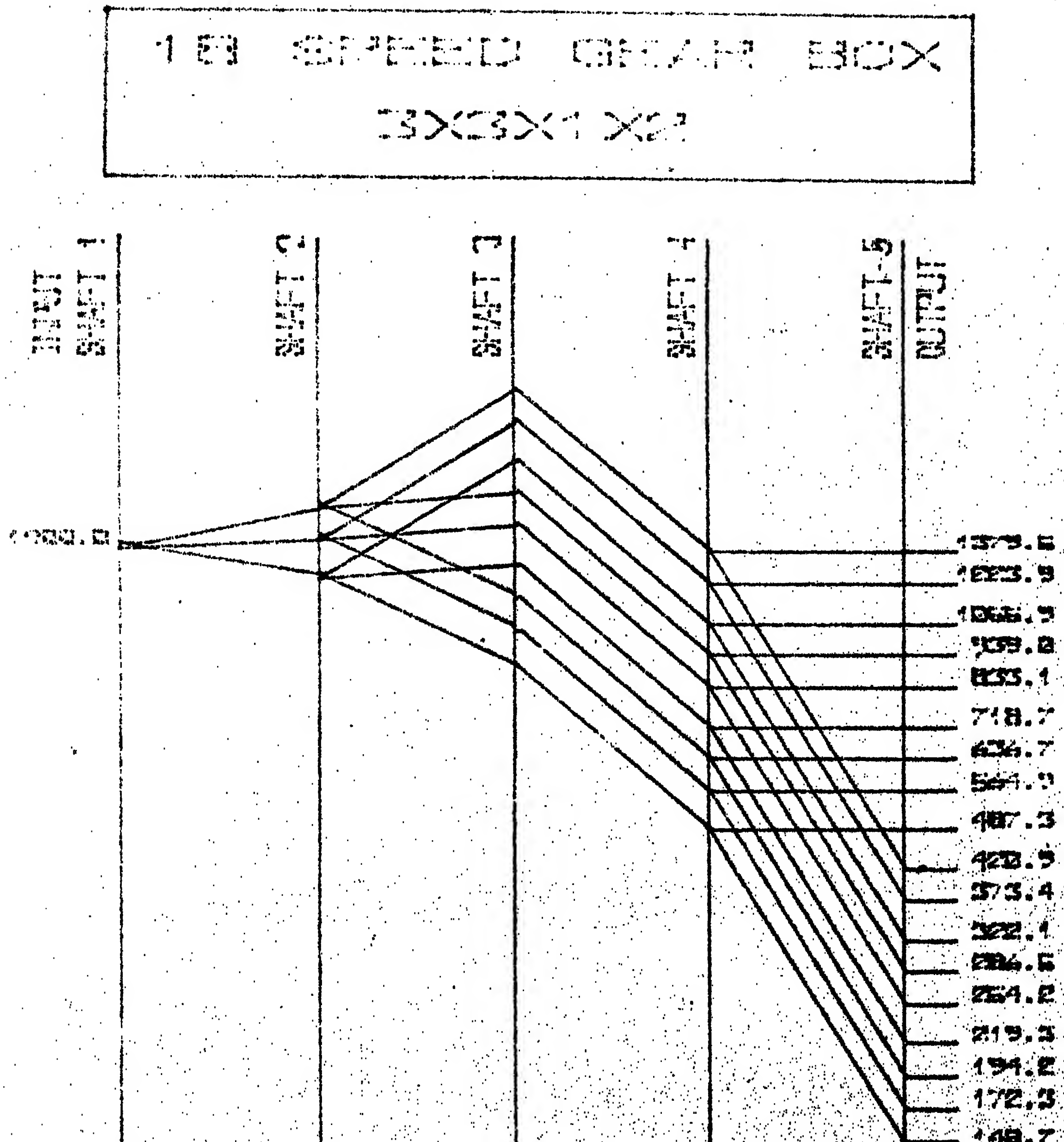
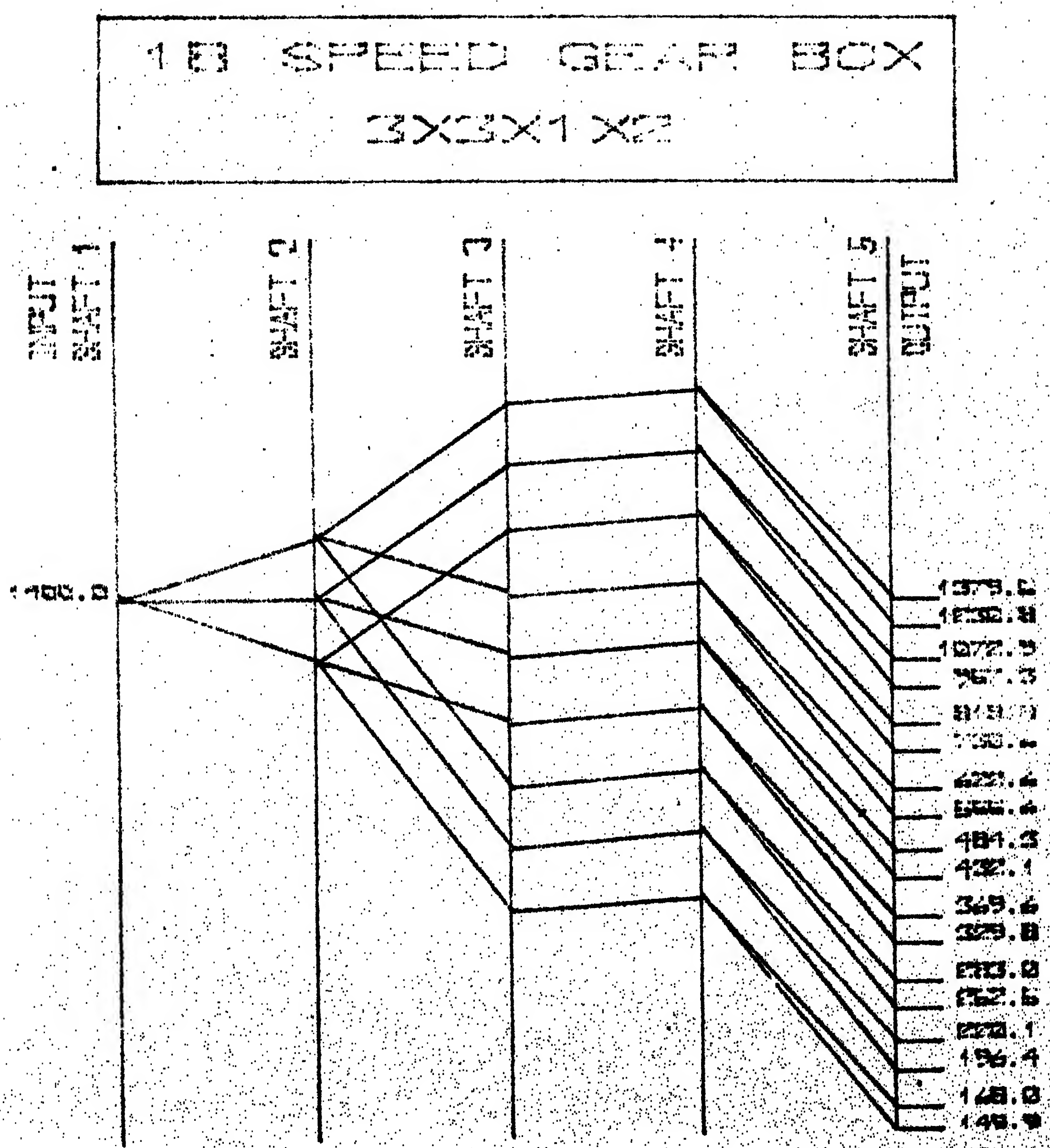

18 SPEED GEAR BOX
3X3X1X2
INPUT
SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
SHAFT-5
OUTPUT
1223.9
718.7
487.3
373.4
322.1
264.2
194.2
172.3

FIG. 5·17



18 SPEED GEAR BOX
3X3X1X2
INPUT
SHAFT 1
SHAFT 2
SHAFT 3
SHAFT 4
SHAFT 5
OUTPUT
484.3
432.1
369.6
196.4

FIG 5.18

24 SPEED GEAR BOX

2X3X2X2X1

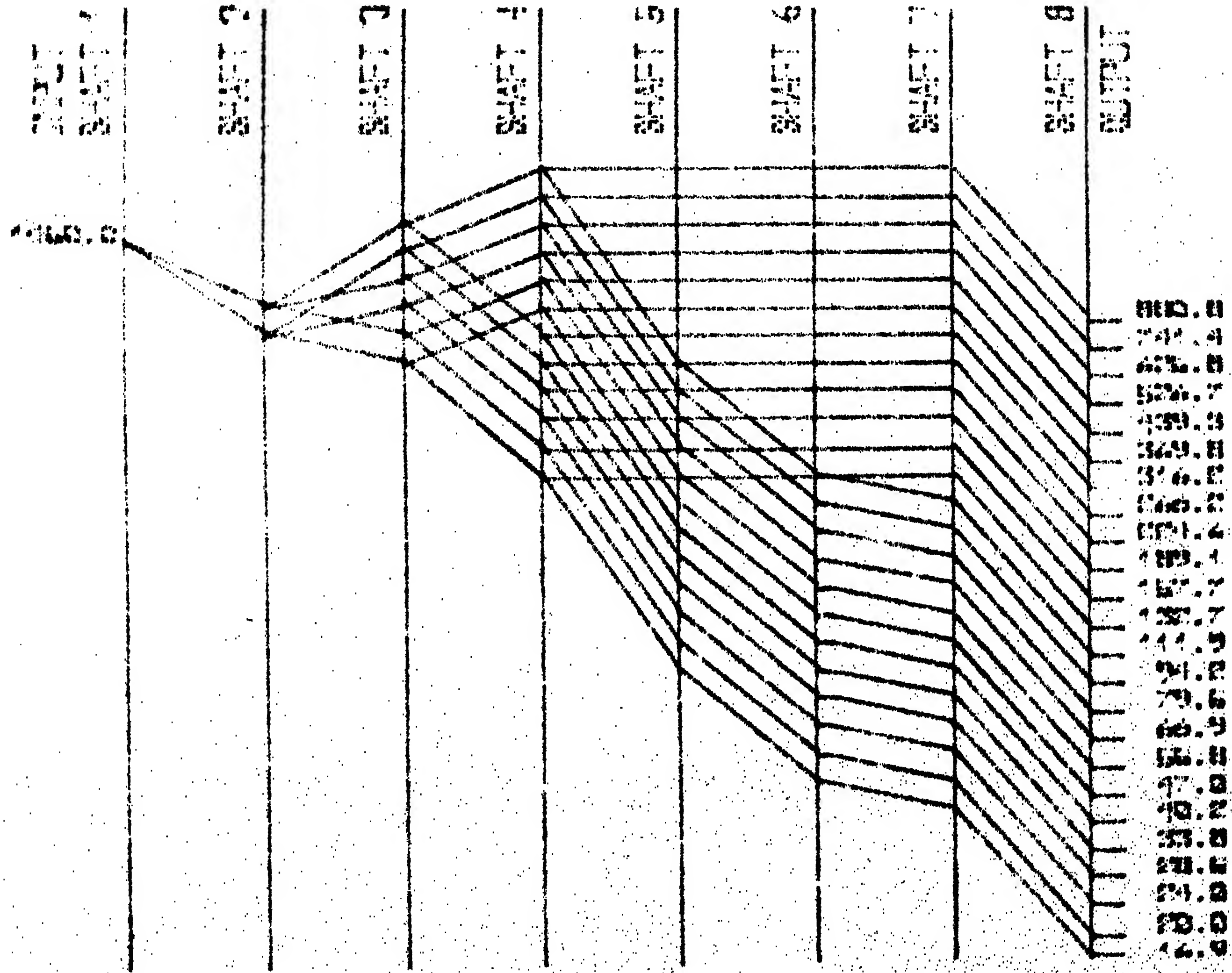


FIG.5.19

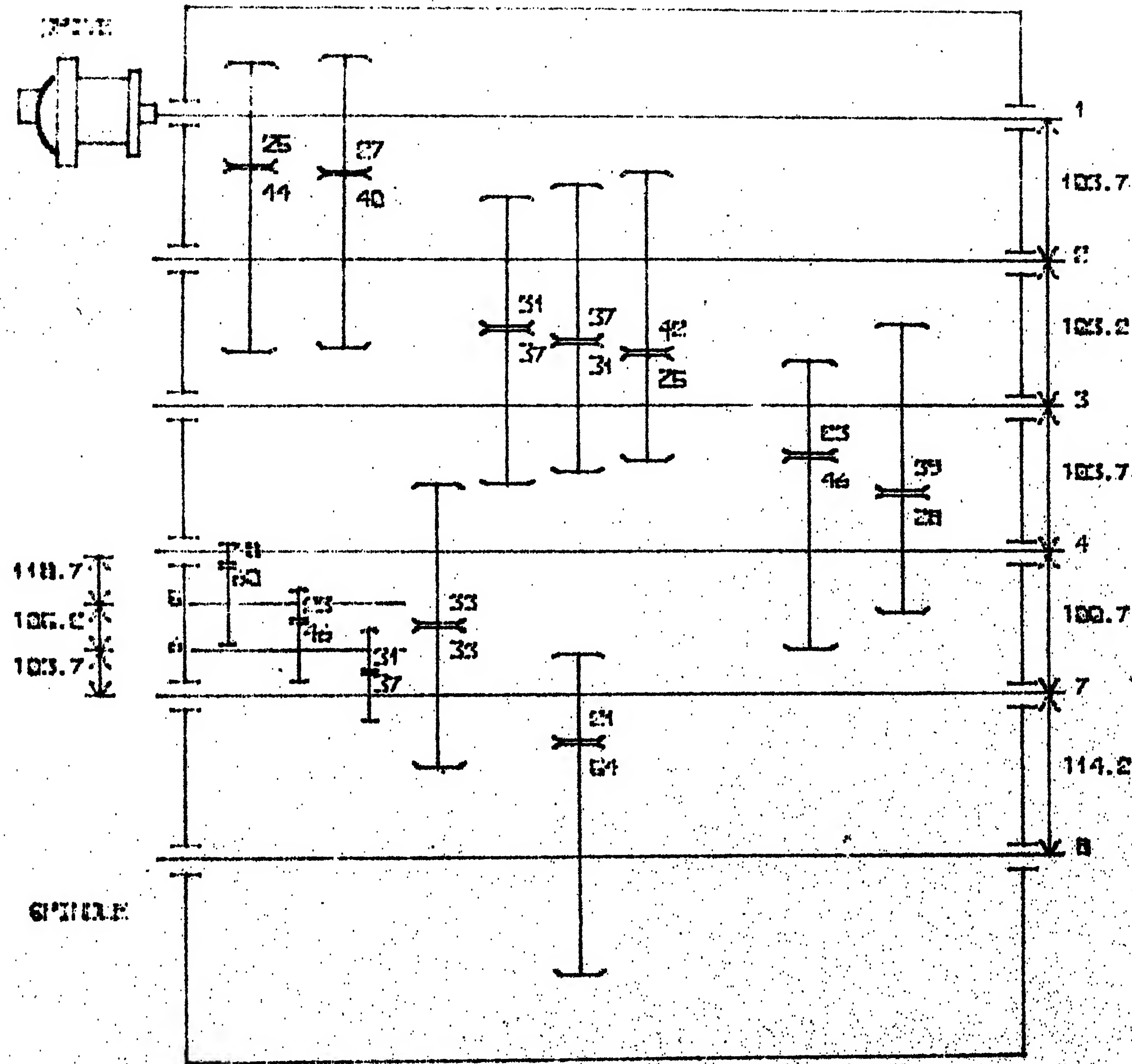


FIG. 5-20

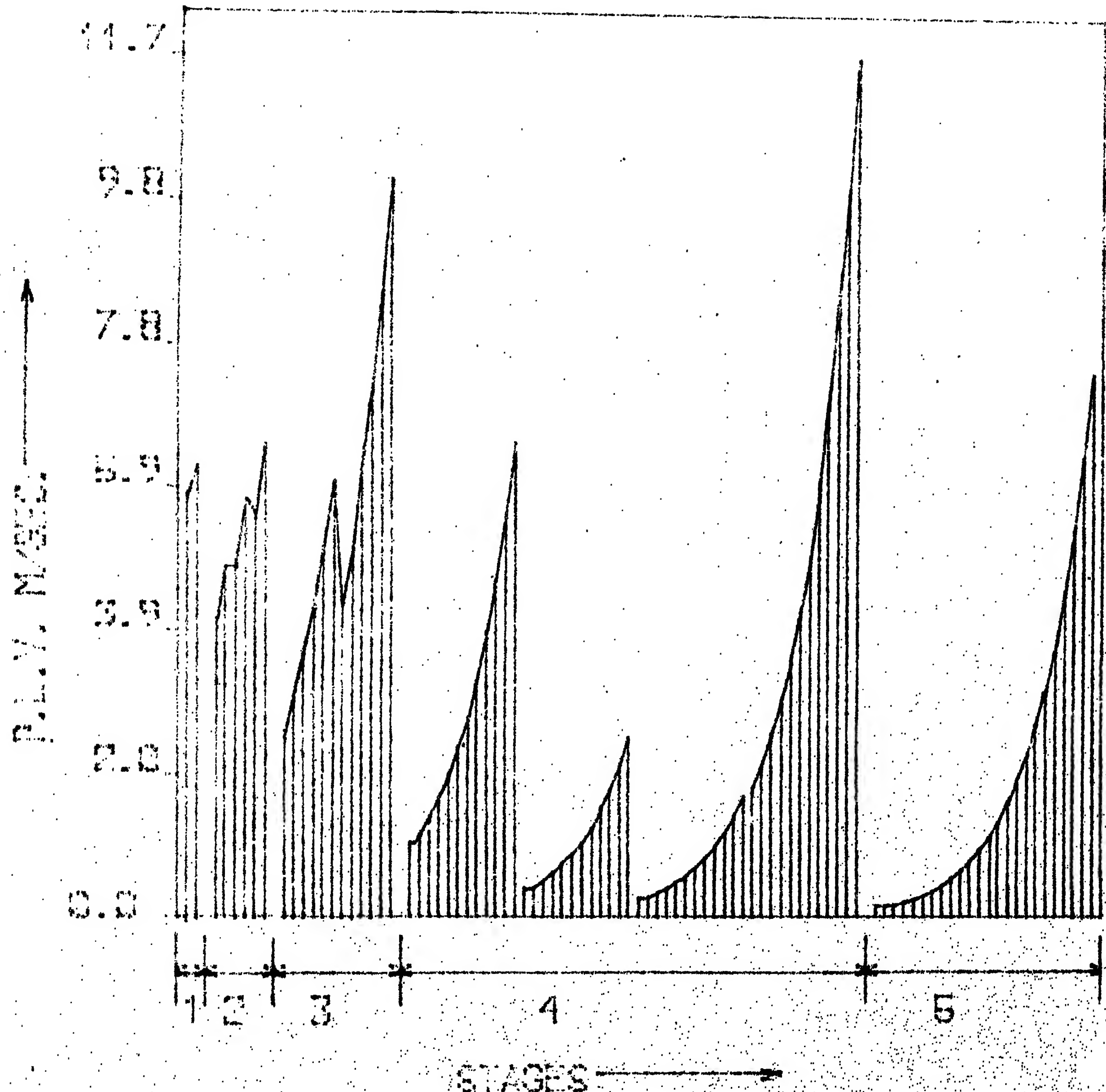


FIG 5.21

TABLE5.1A

TRANSMISSION-RATIOS

STAGE-NO.	RATIO;R	BRANCH-NO.	IDEAL-T.R.	ACTUAL-T.R.	% VARIATION
1	1.18750	1	0.5676	0.5682	0.10494
		2	0.6740	0.6750	0.14709
2	1.18750	1	0.8445	0.8333	-1.31894
		2	1.1896	1.2000	0.76549
		3	1.6768	1.6800	0.04372
3	1.18750	1	0.4993	0.5106	2.27482
		2	1.3974	1.4000	-0.00422
4	1.18750	1	0.1254	0.1229	-1.97618
		2	0.9839	0.9804	-0.54271
5	1.18750	1	0.3908	0.3962	1.38735

TABLE 5.1B

FOR PASSIVE STAGES

STAGE-NO.	RATIO,R	SUB-STAGE NO.	ACTUAL-T.R.
4	1.18750	1	0.2933
		2	0.5000
		3	0.8378

TABLE 5.2A

SPEEDS(RPM) ON ACTIVE SHAFTS

SHAFT-NO.	IDEAL SPEED	ACTUAL SPEED	% VARIATION
1	1450.0	1450.0	0.00000
2	823.0	823.9	0.10494
	977.3	978.7	0.14709
3	695.0	686.6	-1.21539
	825.3	815.6	-1.17380
	980.1	988.6	0.87523
	1163.8	1174.5	0.91770
	1382.0	1384.1	0.14871
	1641.2	1644.3	0.19087
4	347.0	350.6	1.03178
	412.1	416.5	1.07432
	489.3	504.8	3.16996
	581.1	599.7	3.21340
	690.0	706.8	2.42691
	819.4	839.6	2.47003
	973.0	961.2	-1.21956
	1155.5	1141.9	-1.17797
	1372.1	1384.1	0.87097
	1629.4	1644.3	0.91344
	1934.9	1937.7	0.14448
	2297.7	2302.0	0.18664

CONTINUE TO NEXT PAGE

5	43.5	43.1	-0.96480
	51.7	51.2	-0.92310
	61.3	62.0	1.13113
	72.8	73.7	1.17371
	86.5	86.8	0.40275
	102.7	103.2	0.44504
	122.0	118.1	-3.17165
	144.9	140.3	-3.13088
	172.0	170.1	-1.12242
	204.3	202.1	-1.08079
	242.6	238.1	-1.83456
	288.0	282.9	-1.79323
	342.1	344.9	0.84356
	406.2	409.8	0.84602
	482.3	496.7	2.97776
	572.8	590.1	3.02112
	680.2	695.4	2.23609
	807.7	826.1	2.27914
	959.2	945.7	-1.40358
	1139.0	1123.5	-1.36207
	1352.6	1361.8	0.68305
	1606.2	1617.8	0.72545
	1907.3	1906.5	-0.04209
	2265.0	2265.0	0.00000
6	17.0	17.1	0.40917
	20.2	20.3	0.45144
	24.0	24.6	2.53417
	28.5	29.2	2.57734
	33.8	34.4	1.79570
	40.1	40.9	1.83856
	47.7	46.8	-1.82830
	56.6	55.6	-1.78696
	67.2	67.4	0.24935
	79.8	80.1	0.29156
	94.8	94.3	-0.47267
	112.6	112.1	-0.43076
	133.7	136.7	2.24261
	158.7	162.4	2.28566
	188.5	196.8	4.40642
	223.8	233.8	4.45038
	265.8	275.5	3.65446
	315.7	327.3	3.69810
	374.8	374.7	-0.03571
	445.1	445.2	0.00638
	528.6	539.6	2.07988
	627.7	641.0	2.12286
	745.4	755.4	1.34468
	885.2	897.4	1.38735

TABLE 5.28

SPEEDS(RPM) ON PASSIVE SHAFTS

SHAFT NO.	SPEED
1	102.8
	122.2
	148.1
	175.9
	207.3
	246.3
	281.9
	335.0
	406.0
	482.3
	568.4
	675.3
2	51.4
	61.1
	74.0
	88.0
	103.7
	123.1
	141.0
	167.5
	203.0
	241.2
	284.2
	337.6

TABLE 5.3

PITCH LINE VELOCITIES

STAGE-NO.	NO. OF TEETH ON MESHING PAIR	ACTUAL PLV IN M/SEC.	MAX. PLV IN M/SEC.
1	25 - 44	5.6931	
	27 - 40	6.1485	6.1485
2	25 - 30	3.2347 3.8428	
	30 - 25	3.8816 4.6114	
	42 - 25	4.5286 5.3799	5.3799
3	24 - 47	2.5878 3.0743 3.7264 4.4269 5.2169 6.1977	
	42 - 30	4.5286 5.3799 6.5211 7.7471 9.1296 10.8460	10.8460

CONTINUE TO NEXT PAGE

.	* 22 - 75	1.2113 1.4390 1.7443 2.0722 2.4420 2.9010 3.3210 3.9453 4.7822 5.6812 6.6950 7.9537	
	* 23 - 46	0.3715 0.4413 0.5349 0.6355 0.7489 0.8897 1.0184 1.2099 1.4665 1.7422 2.0531 2.4391	
	* 31 - 37	0.2503 0.2974 0.3605 0.4283 0.5047 0.5996 0.6863 0.8154 0.9883 1.1741 1.3836 1.6438	

CONTINUE TO NEXT PAGE

	50 - 51	2.3023 2.7352 3.3154 3.9386 4.6415 5.5141 6.3122 7.4989 9.0896 10.7984 12.7254 15.1178	15.1178
5	21 - 53	0.1532 0.1820 0.2207 0.2621 0.3089 0.3670 0.4201 0.4991 0.6050 0.7187 0.8470 1.0062 1.2270 1.4576 1.7668 2.0990 2.4736 2.9386 3.3639 3.9963 4.8441 5.7547 6.7817 8.0566	8.0566

* PASSIVE TRANSMISSION

TABLE 5.4A

DESIGN OUTPUT FOR MULTI-SPEED GEAR BOX

TOTAL NUMBER OF SPEEDS ON THE SPINDLE= 24

STRUCTURE OF TRANSMISSION TREE= 2 X 3 X 2 X 2 X 1 X

STAGE NO.	RATIO R	GEAR PAIR NO.	MODULE IN MM.	HALIX ANGLE(DEG)	PRESSURE ANGLE(DEG)	DRIVER GEAR TEETH	DRIVEN GEAR TEETH	IDEAL C.D.	CORRECTED C.D.	TOTAL CORRECTION	ACTUAL C.D.
1	1.18750	1	3.00	0.00	20.00	25	44	103.500	105.198	0.600	103.7
		2	3.00	0.00	20.00	27	40	100.500	102.195	0.600	
2	1.18750	1	3.00	0.00	20.00	25	30	82.500	84.349	0.665	84.5
		2	3.00	0.00	20.00	30	25	82.500	84.349	0.665	
		3	2.50	0.00	20.00	42	25	83.750	85.163	0.600	
3	1.18750	1	3.00	0.00	20.00	24	47	106.500	109.700	0.600	109.0
		2	3.00	0.00	20.00	42	30	108.000	109.701	0.600	
4	1.18750	1	*	*	*	*	*	*	*	*	*
		2	2.50	0.00	20.00	50	51	126.250	127.687	0.500	127.7
5	1.18750	1	3.00	22.00	20.00	21	53	119.717	121.438	0.600	121.4

* SEE TABLE-4B DUE TO PASSIVE SHAFT INCLUSION

TABLE 5.4B

FOR PASSIVE STAGES

STAGE NO.	RATIO R	SUB-STAGE NO.	MODULE IN MM.	HALIX ANGLE(DEG)	PRESSURE ANGLE(DEG)	DRIVER GEAR TEETH	DRIVEN GEAR TEETH	IDEAL C.D.	CORRECTED C.D.	TOTAL CORRECTION	ACTUAL C.D.
4	1.18750	1	3.00	0.00	20.00	22	75	145.500	147.222	0.600	147.7
		2	3.00	0.00	20.00	23	46	103.500	105.198	0.600	105.2
		3	3.00	0.00	20.00	31	37	102.000	103.697	0.500	103.7

TABLE 5.5

INSPECTION DATA OF GEARS (DIM. IN MM.)

STAGE NO.	BRANCH NO.	TOTAL CORR.	NO. OF TEETH	PROFILE SHIFT	PITCH CIR. DIA.	TIP CIR. DIA.	BASE CIR. DIA.	ROOT CIR. DIA.	ROLLING CIR. DIA.	ROLLER DIA.	OVER ROLLER READING MINIMUM	OVER ROLLER READING MAXIMUM	FACE WIDTH
1	1	0.067	25	0.123	75.000	81.737	70.477	68.540	75.145	4.9	78.876	78.93	30.00
			44	-0.056	132.000	137.660	124.039	124.463	132.255	5.0	134.617	134.676	
	2	1.183	27	0.583	81.000	89.405	76.115	77.301	83.579	4.9	87.543	87.588	30.00
			40	0.599	120.000	128.899	112.763	116.395	123.821	4.9	126.808	126.853	
2	1	0.723	25	0.368	75.000	82.871	70.477	70.010	76.818	4.9	80.365	80.411	30.00
			30	0.355	90.000	97.790	84.572	84.929	92.182	4.9	95.439	95.487	
	2	0.723	30	0.355	90.000	97.790	84.572	84.929	92.182	4.9	95.439	95.487	30.00
			25	0.368	75.000	82.871	70.477	70.010	76.916	4.9	80.365	80.411	
	3	0.310	42	0.099	105.000	110.444	98.666	99.493	105.940	4.1	108.164	108.217	25.00
			25	0.211	62.500	68.507	58.731	57.556	63.060	4.1	66.130	66.182	
3	1	0.902	24	0.460	72.000	80.352	67.656	67.563	73.590	4.9	78.023	78.067	30.00
			47	0.441	141.000	149.237	132.497	136.448	144.310	5.0	146.682	146.730	
	2	0.345	42	0.137	126.000	132.752	118.401	119.620	127.167	4.9	130.122	130.175	30.00
			30	0.208	90.000	97.180	84.572	84.048	90.833	4.9	94.566	94.617	
4	1 *	0.600	22	0.373	66.000	74.083	62.020	61.038	66.781	4.9	71.538	71.585	30.00
			75	0.227	225.000	232.206	211.431	219.162	227.663	5.0	229.434	229.445	
	*	0.600	23	0.339	69.000	76.830	64.839	63.635	70.132	4.9	74.165	74.212	35.00
			46	0.261	138.000	145.361	129.678	132.365	140.264	5.0	142.691	142.742	
	*	0.590	31	0.310	93.000	100.655	87.391	87.562	94.547	4.9	98.061	98.110	50.77
			37	0.290	111.000	118.531	104.306	105.538	112.846	4.9	115.959	116.008	
	2	0.604	50	0.303	125.000	131.396	117.462	120.517	126.435	4.1	129.219	129.268	25.00
			51	0.301	127.500	133.883	119.611	123.364	128.964	4.1	131.645	131.695	
5	1	0.585	21	0.347	67.948	75.882	63.249	62.829	68.903	4.9	73.185	73.235	129.53
			53	0.239	171.487	178.771	159.626	165.718	173.937	5.0	175.931	175.946	

* PASSIVE TRANSMISSION

CHAPTER-6

CONCLUSION

6.1 Technical Summary

The present work is an attempt to develop an interactive graphical design package for multi-speed gearboxes. The program can be executed on a direct view storage tube-type graphics terminal connected to a main-frame computer.

In practice, the entire process of designing a gearbox consists of, designing the kinematic structure of the gearbox followed by the design of individual gears of the gearbox, along with the specifications of the inspection data. When this process is carried out manually it is difficult to explore all the alternative design strategies, since each alternative needs complete analysis which in turn is very elaborate and time consuming.

The salient feature of the program developed is, that a designer can quickly explore all alternative design strategies and select the appropriate design. For this purpose the program offers several interactive features by which the input data can be modified and the entire design process can be repeated. In short, the designer

can select not only simply a feasible solution, but can select through iterations a better feasible solution.

The results of the design can be displayed in a graphical form, such as the speed diagram and the line diagram. This enables the designer to review the design in a quick, comprehensive manner. Once approved, the design details can be displayed in a tabular form which can also be reviewed quickly.

The present version of the program calculates the face widths of gears using standard approach recommended in the DIN standards specifications. This may not be the practice followed by all the designers. Hence it is necessary to incorporate other standard specifications such as, the IS-Codes or the AGMA-Codes and then ask the designer to exercise his option.

6.2 Recommendations for Further Work

The present version of the program developed does not deal with the design of shafts, bearings and housing. It is necessary for a designer to review the line diagram of the gearbox as developed by the program and then decide about the layout of the gearbox. Layout design is dependent on many constraints of assembly and ease of operation. It is therefore expected that the output of the program developed can be used by the designer in developing

manually, the layout of the gear pairs and the location of support bearings. Alternatively this work can also be accomplished in an interactive graphical mode using a tablet or similar such input device.

Once the locations of gears and the supports has been finalized it is necessary to design all the shafts supporting the gears. The diameters of these shafts should be such that the deflections as well as the stresses induced at critical points in the shaft should be within permissible limits.

Once the sizes of the shafts are finalized then it is necessary to check that the operating speeds of the shafts do not coincide with the critical speeds of the shafts.

It is also necessary to safe-guard the shafts from undesired effects due to whirling.

The design of bearings at all the support location is also an important phase of the design process. The designer has to make a choice between a hydrodynamic bearing and an anti-friction bearing. Once the choice is made then the designer has to perform the necessary design calculations so as to select or to size the appropriate type of bearing.

Finally the designer needs to work out the size and the shape of the casing in which the entire gearbox is housed. This phase is generally heuristic in nature and is based on standard industrial practices.

It is hoped that the enhanced version of the design package should take into account the design of shafts and bearings.

REFERENCES

[1] Sen, G.C. and Bhattacharyya, A., '<u>Principles of Machine Tools</u>', New Central Book Agency, India, 1975, PP. 94-194.

[2] Acherkan, N., '<u>Machine Tool Design</u>', MIR Publishers, Moscow, 1973, Vol. 2.

[3] Niemann, G., '<u>Machine Elements II</u>', Springer International Student Edition, 1979, PP. 3-137.

[4] Darle W. Dudley, '<u>Practical Gear Design</u>', McGraw-Hill Book Company, 1954.

[5] Merritt, H.E., '<u>Gear Engineering</u>', Pitman Publishing , 1971.

[6] White, G., 'Four Speed Gearbox with Six Gears', <u>International Journal of Machine Tool Design and Research</u>, Vol. 3, 1964.

[7] White, G., and Sanger, D.J., 'Design procedure and Computed Solution for a Nine-Speed Gear Train Employing Ten Gears', '<u>International Journal of Machine Tool Design and Research</u>, Vol. 8, 1968, PP. 141-157.

[8] Cinader, F.A., ' A Mathematical Programming Approach to Design of a Transmission', <u>M.S. Thesis, Division of Solid Mechanics, Structures and Mechanics Design</u>, Case Western Reserve University, July 1970.

[9] Singh, P.N., Chakraborty, J.,'A Nonlinear Programming Approach to Optimal Design of Machine Tool Gearbox', <u>ASME Proceedings 72-WA/DE-5</u>, Nov. 1972.

[10] Nanda, D.M.S.,'Computer-Aided Design of Marine Reduction Gearing', <u>M. Tech. Thesis, Department of Mechanical Engineering</u>, I.I.T.,Kanpur, August 1971.

[11] Grover, O.P., 'Computer-Aided Design of an Automotive Epicyclic Gearbox', <u>M.Tech. Thesis, Department of Mechanical Engineering</u>,I.I.T., Kanpur, June 1971.

[12] Fox, R.L., 'Optimization Methods for Engineering Design', Addison-Wesley Pub. Co., 1971

[13] Hadley, G., 'Non Linear Programming', Addison Wesley Publication.

[14] Thornhill, B.R., 'Engineering Graphics and Numerical Control', McGraw Hill

[15] Koenigsberger, F.,' Design Principles of Metal-Cutting Machine Tools', Macmillan Company, New York, 1964, PP. 69-95.

[16] Chornov, N., ' Machine Tools', MIR Publishers, Moscow, 1979, PP. 15-40.

[17] DIN standards 3960-3967, 3992-3995.

[18] Newman, W.M. and Sproull, R.F., 'Principles of Interactive Computer Graphics', McGraw-Hill, Int. Student Edition, Second Edition, 1979.

[19] Mudur, S.P., 'General Purpose Graphics System-User's Manual, NCSDCT Publication, TIFR, Bombay, December, 1975.

APPENDIX-I

I.1 Input Data File for 6-Speed Gearbox

PLEASE GIVE THE NUMBER OF STAGES REQUIRED:(NSTG): 3

PLEASE TYPE THE NUMBER OF SONS BRANCHING OUT FROM EACH NODE IN EVERY STAGE STARTING FROM FIRST STAGE :(NSON) : 1 3 2

PLEASE TYPE THE SPINDLE SPEED RATIO: 1.2500

IS THE SPEED RATIO FOR ALL THE SHAFTS SAME?TYPE Y/N: Y

IS THERE ANY DIS-CONTINUITY IN SPEED DISTRIBUTION ON ANY SHAFT > N

SPECIFY THE SPEED OF MOTOR :(SPMTR): 1000.0000

SPECIFY THE POWER OF INPUT MOTOR IN KW.:(POWMOT): 5.00000

PLEASE TYPE THE LOWEST SPEEDS AT EVERY SHAFT, STARTING FROM SECOND:(SPLOW): 398.00 315.00 160.00

PLEASE GIVE THE VALUE OF MINIMUM NUMBER OF TEETH PERMISSIBLE:(NZMIN): 18

PLEASE GIVE THE VALUE OF TRANSMISSION RATIO CORRESPONDING TO MINIMUM NUMBER OF TEETH:(VTRABS): 3.500

DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS? TYPE Y/N: Y

PLEASE GIVE THE VALUE OF MODULE IN MM.: 3.00

PLEASE TYPE THE VALUE OF PRESSURE ANGLE:(ALFASO): 20.00

DO YOU WANT TO SPECIFY HELIX ANGLE?TYPE Y/N: N

IN STAGE 1 MINIMUM NUMBER OF TEETH CALCULATED ARE 21 CORRESPONDING TO T.R. .3980

DO YOU WANT TO CHANGE THIS CHOICE? TYPE Y/N: N

IN STAGE 2 MINIMUM NUMBER OF TEETH CALCULATED ARE 30 CORRESPONDING TO T.R. .7950

```
DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

IN STAGE 3 MINIMUM NUMBER OF TEETH CALCULATED ARE
24 CORRESPONDING TO T.R.   .5115

DO YOU WANT TO CHANGE THIS CHOICE TYPE? Y/N:  N

OPTION FOR PROFILE SHIFTING:--

 (1) FOR LARGE CONTACT RATIO--(a)MARGINALLY LARGE     P1
                              (b)MODERATELY LARGE     P2
                              (c)HIGHLY LARGE         P3
 (2) FOR BALANCED TEETH     --(a)MARGINALLY BALANCED  P4
                              (b)MODERATELY BALANCED  P5
                              (c) HIGHLY BALANCED     P6

 (3) FOR HIGH ROOT AND      --(a)MARGINALY HIGH       P7
 SURFACE STRENGTH             (b)MODERATELY HIGH      P8
                              (c)HIGHLY LARGE          P9

TYPE THE REQUIRED CURVE NUMBER GIVEN ON
THE EXTREME RIGHT HAND SIDE:                      P6

DO YOU WANT GEAR INSPECTION DATAS? TYPE Y/N:  N

WHICH DIAGRAM YOU WANT TO SEE, RAY/LINE/VELOCITY?
TYPE RAY/LINE/VELO/EXIT TO FINISH  > EXIT
```

I.2 Input Data File for 18-Speed Gearbox

```
PLEASE GIVE THE NUMBER OF STAGES REQUIRED:(NSTG):  4

PLEASE TYPE THE NUMBER OF SONS BRANCHING OUT FROM
EACH NODE IN EVERY STAGE STARTING FROM FIRST STAGE:
(NSON):  3  3  1  2

PLEASE TYPE THE SPINDLE SPEED RATIO:  1.1400
IS THE SPEED RATIO FOR ALL THE SHAFTS SAME?TYPE Y/N:  N

TYPE THE SHAFT NUMBER >  2

TYPE THE VALUE OF RATIO IN TERMS OF POWER INDEX
OF THE SPINDLE SPEED RATIO:  6

ANY MORE RATIO CHANGE LEFT:TYPE Y/N >  N

IS THERE ANY DIS-CONTINUITY IN SPEED DISTRIBUTION
ON ANY SHAFT >  Y

TYPE THE SHAFT NUMBER >  3

TYPE THE VALUE OF DIS-CONTINUITY IN TERMS OF POWER
INDEX OF THE SPINDLE SPEED RATIO:  4

TYPE THE NUMBER OF DIS-CONTINUITIES  >  2

ANY MORE DIS-CONTINUITY LEFT ON ANY OTHER SHAFT?
TYPE Y/N >  Y

TYPE THE SHAFT NUMBER  >  4

TYPE THE VALUE OF DIS-CONTINUITY IN TERMS OF POWER
INDEX OF THE SPINDLE SPEED RATIO:  4

TYPE THE NUMBER OF DIS-CONTINUITIES  >  2

ANY MORE DIS-CONTINUITY LEFT ON ANY OTHER SHAFT?
TYPE  Y/N  >  N

SPECIFY THE SPEED OF MOTOR: (SPMTR): 1400.0000

SPECIFY THE POWER OF INPUT MOTOR IN KW.:
(POWMOT):  5.00000
```

```
PLEASE TYPE THE LOWEST SPEEDS AT EVERY SHAFT,
STARTING FROM SECOND:(SPLOW): 504.00  418.00
410.00   150.00

PLEASE GIVE THE VALUE OF MINIMUM NUMBER OF TEETH
PERMISSIBLE: (NZMIN):  18

PLEASE GIVE THE VALUE OF TRANSMISSION RATIO
CORRESPONDING TO MINIMUM NUMBER OF TEETH:(VTRABS):   3.500

DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS?TYPE Y/N:  Y

PLEASE GIVE THE VALUE OF MODULE IN MM. :  3.00

PLEASE TYPE THE VALUE OF PRESSURE ANGLE:(ALFASO): 20.00

DO YOU WANT TO SPECIFY HELIX ANGLE?TYPE Y/N:  N

IN STAGE 1 MINIMUM NUMBER OF TEETH CALCULATED ARE
20  CORRESPONDING TO T.R.    .3600

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

IN STAGE 2 MINIMUM NUMBER OF TEETH CALCULATED ARE
31 CORRESPONDING TO T.R.    .8360

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

IN STAGE 3 MINIMUM NUMBER OF TEETH CALCULATED ARE
34 CORRESPONDING TO T.R.    .9787

DO YOU WANT TO CHANGE CHOICE? TYPE Y/N:   N

IN STAGE 4 MINIMUM NUMBER OF TEETH CALCULATED ARE
20 CORRESPONDING TO T.R.   .3686

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

OPTION FOR PROFILE SHIFTING:--

 (1) FOR LARGE CONTACT RATIO--(a)MARGINALLY LARGE   P1
                              (b)MODERATELY LARGE   P2
                              (c)HIGHLY LARGE       P3
```

```
(2) FOR BALANCED TEETH      -- (a)MARGINALLY BALANCED   P4
                               (b)MODERATELY BALANCED   P5
                               (c)HIGHLY BALANCED       P6

(3) FOR HIGH ROOT AND
SURFACE STRENGTH            -- (a)MARGINALLY HIGH       P7
                               (b)MODERATELY HIGH       P8
                               (c)HIGHLY LARGE          P9

TYPE THE REQUIRED CURVE NUMBER GIVEN
ON THE EXTREME RIGHT HAND SIDE:          P6

DO YOU WANT GEAR INSPECTION DATAS?TYPE Y/N:  N

WHICH DIAGRAM YOU WANT TO SEE, RAY/LINE/VELOCITY?
TYPE RAY/LINE/VEO/EXIT TO FINISH  >  EXIT
```

I.3 Input Data File for 24-Speed Gearbox

```
PLEASE GIVE THE NUMBER OF STAGES REQUIRED:(NSTG)  5

PLEASE TYPE THE NUMBER OF SONS BRANCHING OUT FROM
EACH NODE IN EVERY  STAGE STARTING FROM FIRST
STAGE: (NSON) : 2 3 2 2 1

PLEASE TYPE THE SPINDLE SPEED RATIO:  1.1875

IS THE SPEED RATIO FOR ALL THE SHAFTS SAME?TYPE Y/N: Y

IS THERE ANY DIS-CONTINUITY IN SPEED DISTRIBUTION
ON ANY SHAFT > N

SPECIFY THE SPEED OF MOTOR:(SPMTR) 1450.0000

SPECIFY THE POWER OF INPUT MOTOR IN KW.:(POWMOT) 5.00000

PLEASE TYPE THE LOWEST SPEEDS AT EVERY SHAFT, STARTING
FROM SECOND:(SPLOW): 823.00 695.00 347.00 43.50 17.00

PLEASE GIVE THE VALUE OF MINIMUM NUMBER OF TEETH
PERMISSIBLE:(NZMIN) :  18

PLEASE GIVE THE VALUE OF TRANSMISSION RATIO CORRESPONDING
TO MINIMUM NUMBER OF TEETH:(VTRABS): 3.500

DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS?TYPE Y/N: N

IN STAGE 1 NUMBER OF GEAR PAIRS ARE 2
DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS OF THIS
STAGE?TYPE Y/N : Y

TYPE THAT VALUE OF MODULE: 3.00

IN STAGE 2 NUMBER OF GEAR PAIRS ARE 3
DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS OF THIS
STAGE?TYPE Y/N: N

TYPE THE DIFFERENT VALUES OF MODULE FOR THIS
STAGE: 3.00 3.00 2.50

IN STAGE 3 NUMBER OF GEAR PAIRS ARE 2
DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS OF THIS
STAGE?TYPE Y/N : Y

TYPE THAT VALUE OF MODULE :  3.00
```

IN STAGE 4 NUMBER OF GEAR PAIRS ARE 2
DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS OF THIS
STAGE TYPE Y/N: Y

TYPE THAT VALUE OF MODULE : 2.50

IN STAGE 5 NUMBER OF GEAR PAIRS ARE 1
DO YOU WANT SAME MODULE FOR ALL GEAR PAIRS OF THIS
STAGE TYPE Y/N : Y

TYPE THAT VALUE OF MODULE : 3.00

PLEASE TYPE THE VALUE OF PRESSURE ANGLE:(ALFASO): 20.00

DO YOU WANT TO SPECIFY HELIX ANGLE TYPE Y/N : Y

SPECIFY THE STAGE, GEAR PAIR HELIX ANGLE: 5 1 22.00

ANYMORE HELIX ANGLE SPECIFICATION LEFT TYPE Y/N : N

IN STAGE 1 MINIMUM NUMBER OF TEETH CALCULATED ARE
25 CORRESPONDING TO T.R. .5676

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N: N

IN STAGE 2 MINIMUM NUMBER OF TEETH CALCULATED ARE
25 CORRESPONDING TO T.R. 1.6768

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N: N

IN STAGE 3 MINIMUM NUMBER OF TEETH CALCULATED ARE
24 CORRESPONDING TO T.R. .5054

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N: N

T.R.IN STAGE 4 EXCEEDS MAXIMUM PERMISSIBLE [3.5]
THEREFORE 1 PASSIVE SHAFT(S) ARE INSERTED IN THIS STAGE,
KEEPING THE LOWEST SPEED(S) ON THEM AS:- 104.34

IS IT ACCEPTABLE TO YOU?TYPE Y OR N: N
TYPE THE NUMBER OF PASSIVE SHAFT(S) YOU WANT TO INSERT 2
PLEASE TYPE THE LOWEST SPEED(S) OF PASSIVE
SHAFT(S): 103.00 51.50

THERE ARE 3 SUBSTAGES, EACH HAVING ONE GEAR PAIR

DO YOU WANT SAME MODULE FOR ALL?TYPE Y/N: Y

PLEASE GIVE THE VALUE OF MODULE IN MM.: 3.00

```
THERE ARE 3 SUBSTAGES, EACH HAVING ONE GEAR PAIR

DO YOU WANT TO SPECIFY HELIX ANGLE?TYPE Y/N: N

IN SUBSTAGE 1 OF STAGE 4 MINIMUM NUMBER OF TEETH
CALCULATED ARE 18 CORRESPONDING TO T.R.  .2938

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  Y

PLEASE TYPE YOUR CHOICE:  22

IN SUBSTAGE 2 OF STAGE 4 MINIMUM NUMBER OF TEETH
CALCULATED ARE 23 CORRESPONDING TO T.R.  .5008

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

IN SUBSTAGE 3 OF STAGE 4 MINIMUM NUMBER OF TEETH
CALCULATED ARE 31 CORRESPONDING TO T.R.  .8460

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

IN STAGE 4 MINIMUM NUMBER OF TEETH CALCULATED ARE 33
CORRESPONDING TO T.R.  .9839

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  Y

PLEASE TYPE YOUR CHOICE:  50

IN STAGE 5 MINIMUM NUMBER OF TEETH CALCULATED ARE 21
CORRESPONDING TO T.R.  .3946

DO YOU WANT TO CHANGE THIS CHOICE?TYPE Y/N:  N

OPTION FOR PROFILE SHIFTING:--

 (1) FOR LARGE CONTACT RATIO-- (a) MARGINALLY LARGE   P1
                               (b) MODERATELY LARGE   P2
                               (c) HIGHLY LARGE       P3
```

```
(2) FOR BALANCED TEETH        --(a)MARGINALLY BALANCED P4
                                (b)MODERATELY BALANCED P5
                                (c)HIGHLY BALANCED     P6

(3) FOR HIGH ROOT AND
SURFACE STRENGTH              --(a)MARGINALY HIGH      P7
                                (b)MODERATELY HIGH     P8
                                (c)HIGHLY LARGE        P9

TYPE THE REQUIRED CURVE NUMBER GIVEN
ON THE EXTREME RIGHT HAND SIDE:          P6

DO YOU WANT GEAR INSPECTION DATAS?TYPE Y/N:  Y

TO CALCULATE OVER ROLLER READING GIVE WIDTH OVER
TEETH ALLOWANCE, TO BE TAKEN SAME FOR ALL GEARS(i.e.AW).
UPPER ALLOWANCE:  -0.06700

LOWER ALLOWANCE:  -0.05000

DO YOU WANT TO CHANGE ANY FACE WIDTH?TYPE Y/N:  Y

TYPE THE STAGE NO. BRANCH NO.:  5  1

TYPE YOUR CHOICE:  120.50

TYPE "T" TO SEE MODIFIED TABLE, "C" TO CHANGE ANY
OTHER FACE WIDTH, "E" TO END THE EDIT MODE.TYPE YOUR
CHOICE:  E

WHICH DIAGRAM YOU WANT TO SEE, RAY/LINE/VELOCITY?
TYPE RAY/LINE/VELO/EXIT TO FINISH  > EXIT
```